공학석사 학위논문

An Analysis of Engine Performance Between Diesel and Gas Mode of Large, 2-Stroke Marine Propulsion Engines

지도교수 조 권 회

2019年 8月

한국해양대학교 대학원
해양플랜트운영학과
James Flowers

본 논문을 James Flowers의 공학석사 학위논문으로 인준함.

위원장 이 원 주 (印)

위 원 조 권 회 (印)

위 원 이 지 웅 (印)

2019년 06월 11일

한국해양대학교 대학원

Table of Contents

List of Tables

List of Figures

List of Abbreviations

AFR	Air-Fuel Ratio
ATDC	After Top Dead Center
BDC	Bottom Dead Center
BHP	Brake Horsepower
BMEP	Brake Mean Effective Pressure
BOG	Boil-Off Gas
C_2H_6	Ethane
C_3H_8	Propane
CA	Crank Angle
CEAS	Computerized Engine Application System
CH_4	Methane
CI	Compression Ignition
cm	Centimeter
CN	Cetane Number
CO_2	Carbon Dioxide
cSt	Centistokes
DWT	Dead Weight Tonnage
ECA	Environmental Control Area
EEDI	Energy Efficiency Design Index
EGR	Exhaust Gas Recirculation
FMEP	Friction Mean Effective Pressure
g	Grams
GHG	Greenhouse Gas
H_2O	Water
HFO	Heavy Fuel Oil
Hg	Mercury
HHV	Higher Heating Value
IGF code	International Code of Safety for Ships using Gases or other Low-flash point Fuels
IMEP	Indicated Mean Effective Pressure

IMO	International Maritime Organization
ISO	International Organization for Standardization
J	Joules
k	Kilo
LCV	Lower Calorific Value
LFL	Lower Flammability Limit
LHV	Lower Heating Value
LNG	Liquefied Natural Gas
m	Meter
m/m	Mass solute per Mass solution
MARPOL	International Convention on the Prevention of Pollution of Ships
MCR	Maximum Continuous Rating
MDO	Marine Diesel Oil
MEP	Mean Effective Pressure
MEPC	Marine Environment Protection Committee
MGO	Marine Gas Oil
mm	Millimeter
MMBtu	Million British thermal units
mmWC	Water Column (millimeter)
MN	Methane Number
mol	Mole
MT	Metric Ton, Million Tons
MTPA	Million Tons Per Annum
MV	Motor Vessel
N_2	Nitrogen
NCR	Nominal Continuous Rating
NG	Natural Gas
NH_3	Ammonia
NM	Nautical Miles
NOx	Nitrogen Oxides
Pcomp	Compression Pressure
Pi	Mean Indicated Pressure

PM	Particulate Matter
Pmax	Maximum Cylinder Pressure
Pscav	Scavenging air Pressure
RPM	Revolutions Per Minute
SCR	Selective Catalytic Reduction
SDF	Specified Dual-Fuel
SFOC	Specific Fuel Oil Consumption
SGC	Specific Gas Consumption
SI	Spark Ignition
SOx	Sulfur Oxides
TDC	Top Dead Center
UFL	Upper Flammability Limit
ULSFO	Ultra-Low-Sulfur Fuel Oil
USD	United States Dollar
VIT	Variable Injection Timing
v/v	Volume solute per Volume solution
W	Watts

An Analysis of Engine Performance Between Diesel and Gas Mode of Large, 2-Stroke Marine Propulsion Engines

James Flowers

Department of Offshore Plant Management
Graduate School of Korea Maritime and Ocean University
(Academic Advisor: Professor Cho, Kwon Hae)

Abstract

Growing energy demand and increasing environmental awareness have given way to new technologies in the maritime industry. One of these technological advances is the use of liquefied natural gas (LNG) as an alternative fuel to diesel. The MV Ilshin *Green Iris* is South Korea's and the world's first LNG-powered bulk carrier and represents a significant shift towards realizing an era of clean energy. It utilizes a dual-fuel, high-pressure gas injection, low-speed marine engine as its source of propulsion and can operate in either a diesel mode or a gas mode. This paper aims to compare between modes to validate that dual-fuel operation is a safe and efficient alternative method to traditional marine diesel combustion. The MV Ilshin *Green Iris* serves as a portal to confirm that LNG is a safe, economically viable, and efficient fuel through engine performance analysis.

Based on the collected data and calculations, the ME-GI engine aboard the vessel experiences about a 1.91% higher thermal efficiency and an SFOC average difference of about 16.27%, favoring gas mode. Data collected from the PMI measurements onboard prove that LNG combustion is as operationally efficient as

heavy fuel oil (HFO) and that gas mode operation meets the same power requirements to that of traditional marine diesel combustion.

Key Words: Performance analysis, LNG-powered bulk carrier, Dual-fuel engine, Diesel Mode, Gas Mode, MV Ilshin *Green Iris*

An Analysis of Engine Performance Between Diesel and Gas Mode of Large, 2-Stroke Marine Propulsion Engines

James Flowers

해양플랜트운영학과
한국해양대학교 대학원
(지도교수: 조권회)

논문요지

에너지 수요의 증가와 환경문제에 대한 인식변화로 해운산업에 발전된 새로운 기술이 도입되고 있다. 이와 같은 변화에 따라 전통적인 액체연료를 대체하여 액화천연가스를 사용하는 이중연료 엔진이 개발되고, 이를 천연가스운반선뿐만 아니라 산적화물운반선에도 적용하기에 이르렀다. 그의 일환으로 벌크선 중 세계 최초로 건조된 '그린아이리스'호는 5만톤급 규모의 액화천연가스(LNG) 추진선으로, 미래 친환경 선박으로 주목받고 있다. 이 선박은 벙커C유와 LNG를 함께 사용할 수 있는 이중연료 추진엔진을 탑재하여 디젤모드와 가스모드로 추진된다.

본 논문은 고유황 벙커C유 추진엔진에 대한 LNG이중연료추진엔진의 안전성과 운항 효율성을 확보하기 위해서 디젤모드와 가스모드에서 수집된 데이터와 계산을 토대로 성능을 비교 해석하고자 하였다.본 연구는 MV Ilshin Green Iris를 대상으로 위의 두 가지 운전 모드에서 수집된 데이터와 계산을 토대로 출력, 평균유효압력, 열효율, 연료소비율, 배기가스 온도 및 압축압력 등을 비교하였다. 비교 결과 가스 모드는 거의 디젤 모드만큼 효율적이었다. ME-GI 엔진은 가스 모드에서 작동할 때 약 1.91%의 높은 열효율과, 약 16% 낮은 연료

소비율(SFOC)를 보이며, PMI 측정으로부터 수집된 데이터는 LNG 연소가 HFO 만큼 효율적이며 가스 모드 운전이 동일한 동력 요구 사항을 충족시킨다는 것을 보여주었다.

따라서 본 논문은 LNG가 대안 연료로의 전환 시대에, LNG 연료 운전이 전통적인 디젤 연소의 안전하고 효율적인 대체 방법임을 검증하였다.

주제어: 성능 해석, LNG 추진 벌크선, 이중연료 엔진, 디젤 모드, 가스 모드, 일신 그린 아리리스호

Chapter 1: Introduction

1.1 Background

The growing demand for energy is driving up liquid fuel prices, and the utilization of alternative fuels is becoming increasingly attractive. Also, rising environmental awareness has led to stricter emission regulations, especially in the International Maritime Organization (IMO) Emission Control Areas (ECA). In the future, these areas, as well as globally, will be restricted exclusively to ships that fulfill the IMO tier II & III emission regulations. To meet these challenges for tomorrow's maritime and shipping industry, new LNG-powered vessels are needed. The bulk carrier MV Ilshin *Green Iris* is the world's first LNG-powered bulk carrier to have been built.

The MV Ilshin *Green Iris* is South Korea's first and the world's first LNG-powered bulk carrier. The building of this vessel marks a milestone in the movement towards the use of LNG as a conventional fuel for vessel propulsion. The 50,000 deadweight tonnage (DWT) class bulk carrier entered service in February 2018. It was built as a national charter for South Korea's leading steel manufacturer, POSCO, mostly for the transportation of limestone. POSCO also constructed the vessel's LNG fuel tank made of high manganese steel as an alternative to the more common nickel alloy type. The use of LNG as a primary fuel source marks a momentous occasion for the South Korean government. The government has taken advantage of this to front the country's commitment to the development of an LNG bunkering infrastructure. This is important in order to comply with and make way for the IMO Tier II and III emission regulations. The MV Ilshin *Green Iris* will also be used as a kind of prototype for the South Korean government to assess the direction of LNG-related shipbuilding and associated policies.

Presently, the MV Ilshin *Green Iris* has a domestic route operating only within the South Korean coastline. This means that it does not need to adhere to Tier III standards because it does not operate inside of any ECAs. If it were to operate in an ECA, the vessel was built with additional space in the engine room for later installation of an exhaust gas recirculation (EGR) system for NOx reduction and possibly a selective catalytic reduction (SCR) system.

Emissions from prime movers of ships have largely been the primary focus of environmental issues at sea. The IMO ship pollution rules are contained in the International Convention on the Prevention of Pollution of Ships (MARPOL). The IMO emission regulations are typically referred to as Tier I, Ⅱ, and Ⅲ standards. In 1997, the MARPOL Convention was first adopted which included Annex VI, titled “Regulations for the Prevention of Air Pollution from Ships”, and defined the Tier I standards. This Annex enforces limits on nitrogen oxides (NOx) and sulfur oxides (SOx) emissions from ships. Also, it bans intentional emissions of ozone-depleting substances. In 2008, Annex VI amendments introduced the Tier Ⅱ and Ⅲ standards. The MARPOL Annex VI revisions include a drastic reduction in SOx, NOx, and particulate matter (PM) limits from ship emissions. Tier Ⅲ limitations only apply to the emission control areas (ECA) defined by MARPOL. The revised Annex VI amendments came into effect on July 1st, 2010. In addition, the 2011 amendments to MARPOL Annex VI introduced mandatory measures to reduce emissions of greenhouse gases (GHG). Compliance with these revisions is currently determined by periodic inspections and surveys conducted by local governing authorities. Details of the emission regulations are described in the following section.

1.2 Emission Regulations

Currently, there are two sets of emission and fuel quality requirements established by Annex VI: global requirements and requirements pertaining to ECAs. An ECA can limit the amount of SOx, NOx, or PM, or all three types of emissions from ships. Tier I and Ⅱ limits apply globally, while Tier Ⅲ limits only apply to ECAs.

Existing ECAs include [1]:

- Baltic Sea (SOx: adopted 1997 / enforced in 2005; NOx: 2016/2021)
- North Sea (SOx: 2005/2006; NOx: 2016/2021)
- North America, includes most of US and Canadian coast (NOx & SOx: 2010/2012)
- US Caribbean, includes Puerto Rico and US Virgin Islands (NOx & SOx: 2011/2014)

The NOx emission limits are for diesel engines and depend on the engine maximum operating speed (n, rpm) as shown in **Table 1.1** and **Fig. 1.1**.

Table 1.1 MARPOL Annex VI NOx Emission Limits

Tier	Date	NOx Limit, g/kW·h		
		n < 130	130 ≤ n < 2000	n ≥ 2000
Tier I	2000	17.0	$45 \cdot n^{-0.2}$	9.8
Tier Ⅱ	2011	14.4	$44 \cdot n^{-0.23}$	7.7
Tier Ⅲ	2016	3.4	$9 \cdot n^{-0.2}$	1.96

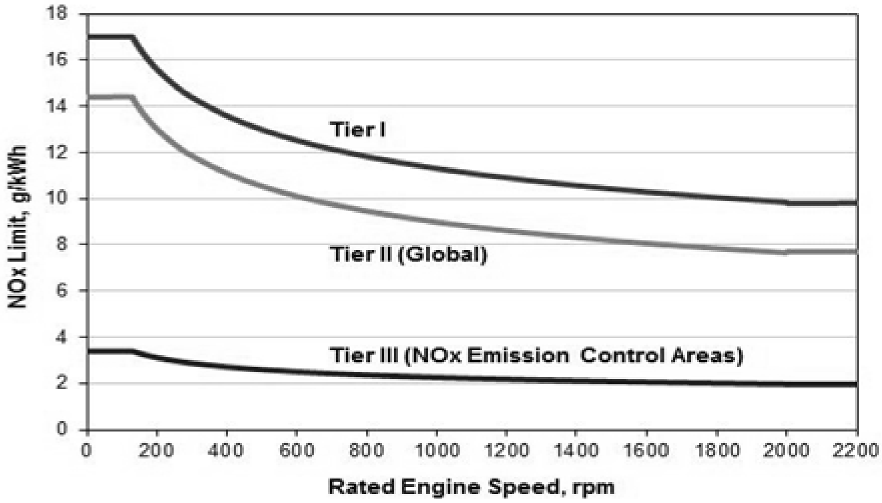


Fig. 1.1 MARPOL Annex VI NOx Emission Limits [2]

The IMO expects tier Ⅱ compliance through combustion process optimization. The considerations examined by engine manufacturers include fuel injection pressure and timing, fuel nozzle flow area, cylinder compression volume, and exhaust valve timing. Tier Ⅲ compliance is expected to be met through NOx emission control technologies. These include but are not limited to water injection into the combustion process, SCR, and EGR systems [2].

SOx emission restrictions are expected to be met through limiting the sulfur content of the fuel used. These limitations are shown in **Table 1.2** and **Fig. 1.2**. The concentration of the sulfur in the fuel is expressed as a mass/mass percent (mass solute per mass solution, % m/m).

Table 1.2 MARPOL Annex VI Fuel Sulfur Limits

<table>
<tr><th rowspan="2">Date</th><th colspan="2">Sulfur Limit in Fuel (% m/m)</th></tr>
<tr><th>SOx ECA</th><th>Global</th></tr>
<tr><td>2000</td><td>1.5%</td><td rowspan="2">4.5%</td></tr>
<tr><td>2010</td><td rowspan="2">1.0%</td></tr>
<tr><td>2012</td><td rowspan="2">3.5%</td></tr>
<tr><td>2015</td><td rowspan="2">0.1%</td></tr>
<tr><td>2020</td><td>0.5%</td></tr>
</table>

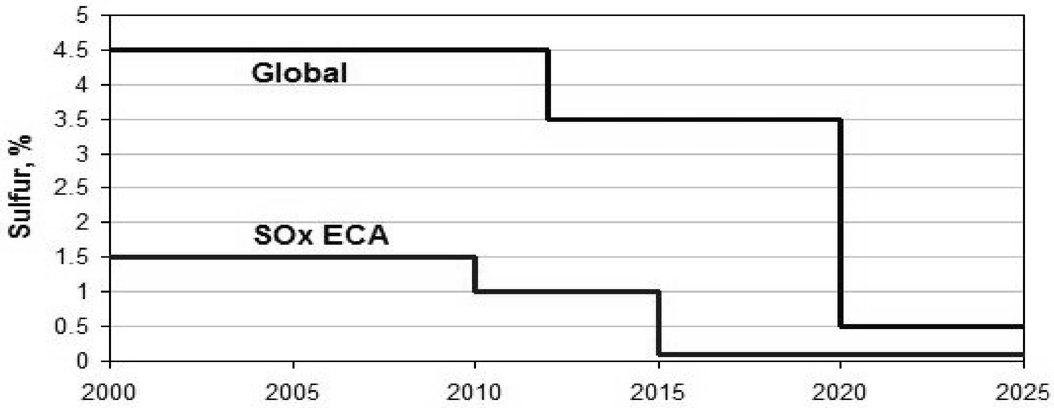


Fig. 1.2 MARPOL Annex VI Fuel Sulfur Limits [2]

Currently, there is no mandate requiring the use of distillate fuels. For example, a residual fuel like HFO is allowed provided that it meets the sulfur limit. Alternative SOx emission reduction technologies, such as scrubbers, are permissible in both SOx ECAs and globally. As can be seen from **Fig. 1.2**, there is a dramatic decrease in the sulfur content limit in fuel from 3.5% to 0.5% that will be enforced in 2020. This will be very problematic for shipping companies with vessels that have no space to install scrubbers or exhaust gas cleaning systems and are currently using fuels with a sulfur content above 0.5%.

Fuel combustion emits GHG in the form of CO_2. Depending on the carbon content of the fuel used, different fuels produce different amounts of CO_2 relative to the energy they produce when burned. The heat content produced when a fuel burns is primarily determined by its carbon and hydrogen content. LNG, having a very low carbon content, produces little amounts of CO_2 when burned. This makes LNG a much more environmentally friendly fuel when compared to today's widely used diesel fuels. **Table 1.3** shows reaction formulas for LNG, marine gas oil (MGO), and HFO along with their respective post-combustion fuel to CO_2 ratio. As can be seen, LNG has the lowest fuel to CO_2 ratio per kilogram of fuel.

Table 1.3 Fuel Type Reaction Formula and Fuel to CO_2 Ratio

Fuel	Reaction formula	Fuel to CO_2 (kgf)
LNG	$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$	1 : 2.75
MGO	$C_{12}H_{26} + 18½O_2 \rightarrow 12CO_2 + 13H_2O$	1 : 3.10
HFO	$C_{18}H_{38} + 27½O_2 \rightarrow 18CO_2 + 19H_2O$	1 : 3.12

The IMO has recently agreed on a limit to the amount of GHG from ship emissions. This agreement was drawn by the Marine Environment Protection Committee (MEPC) in April of 2018 and requires the shipping sector's annual GHG output to be cut by at least 50% by 2050. Additionally, it requires reducing CO_2 emissions per transport work, on average, by at least 70% by 2050. This new target for strict GHG emission limitations makes LNG as an alternative fuel ever more attractive. Additionally, the MEPC has agreed to begin emission reductions and pursue carbon-free emissions entirely. Current technology will not work for this goal, however, LNG-powered ships may act as a stepping stone towards realizing this goal [3].

1.3 Purpose of Thesis

Complying with IMO emission regulations pose enormous obstacles for the maritime industry. Therefore, LNG as a fuel used by prime movers on ships is becoming an increasingly attractive option for ship owners. The utilization of LNG in marine diesel engines, however, poses many technological challenges. Proving to be a safe technology, dual-fuel engines using LNG provide a clean, robust, and efficient method of propulsion. To accomplish this, dual-fuel engines must utilize one of three modes: diesel mode, gas mode, or an intermediary mode. To avoid redundancy this paper will focus on the former two modes: gas mode and diesel mode. We compare the two modes to ratify the use of LNG as a safe, efficient method of marine propulsion that complies with emerging environmental regulations. Additionally, the challenges of transitioning between these modes will be discussed.

Compliance with maritime emissions regulations typically occurs through one of three pathways. The primary solution is to change the inner workings and means by which the prime movers themselves function. This method, however, is only a temporary solution because as energy demands rise so must engine efficiency. This means technology must be continuously improved in a never-ending cycle. The secondary method is to alter or change measures post-combustion. Utilizing technologies like EGR or SCR are ways to comply with emission regulations via post-combustion. However, these mitigation solutions may be less economically efficient compared to fuel switching and do not target the cause of why the emission regulations exist. The best solution to ultimately comply with these strict, continually changing regulations is merely to change the fuel. This thesis will highlight this solution through a diesel mode and gas mode comparison of the ME-GI engine currently onboard the MV Ilshin *Green Iris*. Factory tests (shop tests) and sea-trial testing data will be analyzed.

Based on current emission regulations and their respective closing limit gaps, all future ships must be constructed in such a way to comply with all of the IMO regulatory requirements. The MV Ilshin *Green Iris* is a prime example of how other shipping companies can meet these standards. The application of alternative fuel, such as LNG, is a sort of icon of what lies ahead for the shipping industry's future.

Relative literature and studies will be presented and discussed to inform the reader of technical information in regards to the purpose of this paper. The author's research data and methodology for evaluating the safety and effectiveness of LNG fuel combustion in large, 2-stroke marine propulsion engines will be presented to validate the aim of the thesis. Finally, a conclusion will be presented to summarize findings.

Chapter 2: Relevant Literature and Research

It must be understood that an alternative fuel, like LNG, must meet four necessary criteria in order to be a suitable choice to replace conventional liquid fuel like MGO, marine diesel oil (MDO), and HFO. Firstly, it must be a very safe and reliable fuel. Secondly, it must be an economically feasible choice. Thirdly, its availability must be high. Lastly, it should produce emissions that meet IMO regulatory requirements. These four criteria are discussed in detail in the following sections as well as the basic operating principles of dual-fuel engines, fuel property comparisons, combustion characterization, and the challenges of mode transitioning.

2.1 Mode Operation

The 6G50ME-C9.5-GI-TⅡ engine onboard the MV Ilshin *Green Iris* is a 2-stroke, low-speed, high-pressure gas injection, dual-fuel engine. When the ME-GI engine is compared to a standard ME engine design, only a few technical differences exist. One of these technical differences is the ME-GI injection system. The ME-GI engine design is considered to be a high-pressure injection system where gas is supplied to the gas injection valves at 300 bar. During diesel mode operation of a ME-GI engine, the injection pressure is about 800 bar while other common rail systems deliver at around 1,500-2,000 bar. The Wartsila RT-flex, for example, delivers fuel at a pressure of about 1,000 bar.

The ME-GI engine type is a true diesel engine meaning that heat produced by the compression of air is used to ignite the fuel. However, it does require some level of diesel fuel, called pilot fuel, for operation when in dual-fuel mode, or gas mode. During dual-fuel operation, injection of first pilot fuel (to initialize combustion) and then gas fuel into the cylinder is required. There are also different types of valves that are used for the injection of gas and pilot fuel. For both diesel and gas modes the required supporting media is high-pressure gas, fuel oil (pilot oil, HFO), control oil for actuation of gas injection valves, and sealing oil to separate gas and control oil.

With fuel flexibility being the critical advantage for the ME-GI engine, it is essential to understand the limitations of this advantage. The ME-GI engine can operate under three different fuel modes: 1) gas operation mode, or gas mode (with a minimum amount of pilot oil), 2) a specified dual-fuel mode (SDF) with an injection of a fixed gas amount, and 3) fuel-oil-only mode or diesel mode. In gas operation mode the engine can operate with a minimum amount of pilot fuel oil of about 3%. In SDF mode the operator has complete fuel flexibility control with the choice to inject a fixed amount of gas. The main engine control system will then add enough fuel oil until the suitable engine load is reached. In diesel mode, the engine is operating only on fuel oil. In the event of a failure in the gas system, the gas is shut down and the engine returns to diesel mode.

When operating in gas mode, gas operation is possible down to 10% engine load but typically not below 15%. The minimum amount of pilot fuel in gas mode is 3%. **Fig. 2.1** shows the minimum amount of pilot oil required for gas mode operation in relation to engine load. When in dual-fuel mode after mode transitioning, the fuel oil load percent typically remains at 5% with minimal deviation aboard the MV Ilshin *Green Iris* [4].

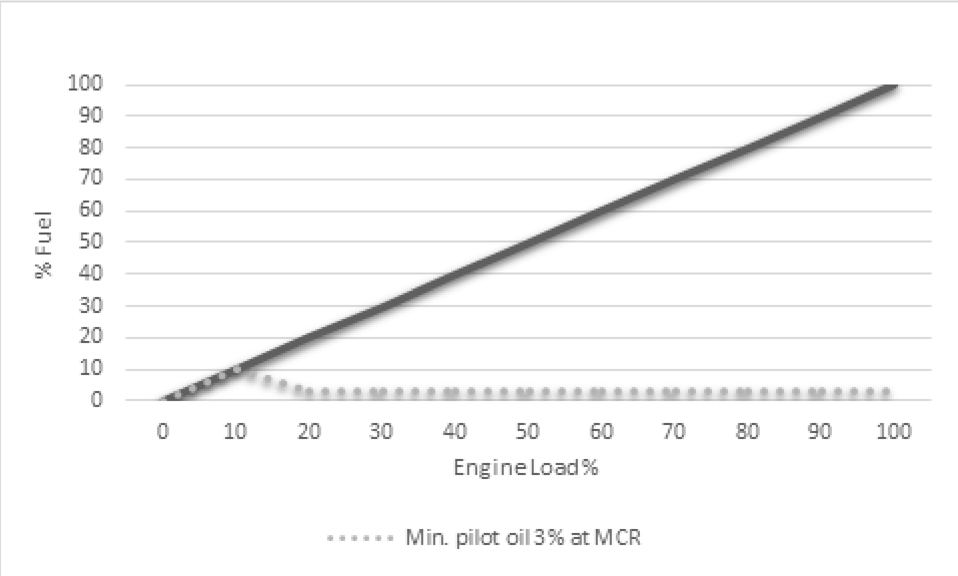


Fig. 2.1 Fuel Index in Gas Operation Mode [4]

Table 2.1 shows approximations of fuel oil consumptions and duration of cruises based on LNG and HFO for the MV Ilshin *Green Iris* operating in gas mode and diesel mode. This information is based on the nominal continuous rating (NCR) of the main engine and when the filling ratio of LNG and MDO is at 90% and 98%, respectively.

Table 2.1 Fuel Oil Consumption and Cruise Duration

Specification	**Gas Mode**	**Diesel Mode**
Daily F/O Consumption (MT, metric tons)	17	21.3
Specific Gravity	0.45	0.9
LCV (kJ/kg)	50,000	42,700
Duration of Cruise (days)	11	16
Cruising Range (NM)	3,600	5,300

In gas mode of an Otto cycle engine, the gas-air mixture is fed into the cylinders during the intake stroke. In the ME-GI dual-fuel propulsion engine of the

MV Ilshin *Green Iris*, however, the engine does not operate in this manner. In both modes the engine operates according to the Diesel cycle and therefore comparing modes is relatively straightforward. Additionally, this makes engine design and control simpler making a seamless transition between modes easier, uncomplicated, and safer.

2.2 Safety of LNG and Fuel Systems

Safety is a factor that must always be addressed first. Liquefied gas sometimes gives an impression of being dangerous. Quite contrary, LNG is one of the safest fuels available in today's market. Historically, LNG has the safest record of any fuel type. Being completely non-toxic, LNG has substantially fewer safety hazards than diesel, gasoline, or other liquid fuels.

LNG is cryogenically cooled in order to reduce its volume. It can be compressed by a factor of about 600 times to make it economically viable for shipping and storage. Because the LNG onboard a ship must utilize cryogenic systems and equipment in order to maintain proper storage temperature there represents a number of safety concerns. If the cryogenic systems were to fail, LNG by itself is still inherently very safe. The reason for this is that LNG has a very high rate of vaporization and leaves behind no residue or pools of liquid, unlike most liquid fuels. If LNG does vaporize the vapor cloud can, however, ignite but only if there is a source of ignition. The following are some common demonstrations used to illustrate the fundamental safety aspects of LNG: Pouring LNG on the ground to show how quickly LNG vaporizes leaving behind no residue, pouring LNG into a container of water and then drinking that water, pouring LNG into a container with live goldfish proving that LNG floats on the surface and does not harm marine life, and extinguishing a cigarette in a container of LNG to show that liquid methane will not ignite because only the vapors are flammable [5].

All of these demonstrations prove that LNG is a very reliable, environmentally friendly, and safe alternative fuel to MDO. Furthermore, LNG storage onboard a ship is very safe. LNG tanks are double walled and very thick making them much stronger than traditional fuel tanks.

Natural gas also has very narrow flammability limits. LNG, comprised primarily of methane (CH_4), has a lower flammability limit (LFL) of about 5% and an upper flammability limit (UFL) of about 15%. **Table 2.2** shows various hydrocarbons and their respective chemical properties for reference.

Table 2.2 Properties of Hydrocarbons [6]

Hydrocarbon	Formula	Molecular Weight	Boiling Point (1atm) (℃)	Flashpoint (℃)	Auto Ignition point(℃)	Flammability Limit (vol %) (LFL/UFL)
Methane	CH_4	16	-161	-	537	5.3/14.0
Ethane	C_2H_6	30	-89	-	510	3.0/12.5
Propane	C_3H_8	44.1	-42	-	467	2.2/9.5
Butane	C_4H_{10}	58.1	-12	-	430	1.9/8.5
Pentane	C_5H_{12}	72.1	28	-51	309	1.4/7.8
Hexane	C_6H_{14}	86.1	50	-29	260	1.2/7.5
Heptane	C_7H_{16}	100.1	80	-18	233	1.2/6.7
Octane	C_8H_{18}	114.1	99	-12	232	1.0/6.0
Nonane	C_9H_{20}	128.1	150	31	285	0.8
Decane	$C_{10}H_{22}$	142.1	174	46	250	0.8
Cyclic Hydrocarbon	C_6H_6	78.1	83	-11	583	1.4
	C_7H_8	92.1	110	4	552	1.4
	C_8H_{10}	106.1	136	17	482	1.1
Inorganic	H_2S	34.1	-60.2	-	260	4.3

This means that if the amount of CH_4 present in the vapor mixture exceeds 15%, the mixture is too rich to burn and if it is lower than 5% the mixture is too lean. When compared to HFO, which has an LFL and a UFL of about 1% and 6%, respectively, LNG does not ignite as readily as a pool of HFO does. The auto-ignition temperature of LNG is also significantly higher than MDO, about 540℃ and 210℃, respectively, meaning that hot surfaces are very unlikely to ignite LNG.

Although potential hazards do exist if LNG is spilled, those hazards are unlikely. In the event that LNG is spilled and a vapor cloud forms and is unable to dissipate there is a chance of fire if there is an ignition source present. LNG can also cause cryogenic burns if spilled and comes into contact with skin. LNG has no smell and is an asphyxiate making it very difficult and dangerous for humans to detect without gas detection.

Despite the few hazards that LNG has, the LNG industry has conducted over 33,000 voyages since 1964 without ever having a substantial spill, any loss of cargo, or environmental incident. The only major incident was in 1944 in Cleveland, Ohio where an LNG tank ruptured spilling over 1 million gallons of LNG into a nearby sewer drain. The vapors leaked into sewer pipes of residential homes ignited and burned down an entire neighborhood killing 128 people. This was a significant lesson learned that created safety standards that are now followed today across many industries [7].

The use of LNG as a fuel is regulated by the International Code of Safety for Ships using Gases or other Low-flashpoint Fuels (IGF Code). Therefore, the ME-GI engine design complies with the IGF Code, however, there are still many safety challenges and a degree of risk involved when operating a dual-fuel engine. Components that are unique to a ME-GI engine that overcome safety concerns and challenges include a chain pipe gas supply system for high-pressure gas distribution, a leakage detection and ventilation system for the double-walled gas supply pipe, a sealing oil system for the gas valves that separates control oil and gas, an inert gas system that enables gas system purging, and a control and safety system for examination of hydrocarbon content of air in the double-walled gas pipes.

The ME-GI dual-fuel engine requires two fuel systems to be maintained: the fuel oil system and fuel gas supply system. The chance of failure in a fuel supply system, when compared to an ordinary ME engine, has now significantly increased. Various factors can influence the operational safety of any engine, however, with more systems added, the risk of failure becomes higher. On the gas supply system of ME-GI engines, gas pipes are designed with double walls. The outer pipe prevents any gas from escaping into the machinery space in the event of a leak or break of the inner gas pipe. Furthermore, the gas pipes are connected to an inert gas purging system composed of 95% nitrogen. This system enables purging of the fuel gas supply system and the gas system on the engine in the event that a leak is detected. With safety challenges overcome and LNG as a relatively safer fuel than most traditional diesel oils, ship owners and shipping companies should have no substantial reason to reject LNG as an alternative fuel [4].

2.3 Economics and Availability of LNG

The second and third criteria that make LNG a respectable alternative fuel is because of its economic feasibility and availability. Global gas demand increases and is expected to grow more rapidly with an increased interest in cleaner energy. **Fig. 2.2** shows the average regional natural gas prices in terms of $/MMBtu from 2010 to January 2019. Most global LNG prices followed an upward trend in 2018, influenced by both rising oil prices and a strong LNG demand in Asia. As new liquefaction capacity is added throughout 2019, however, prices are expected to continue to fall [8].

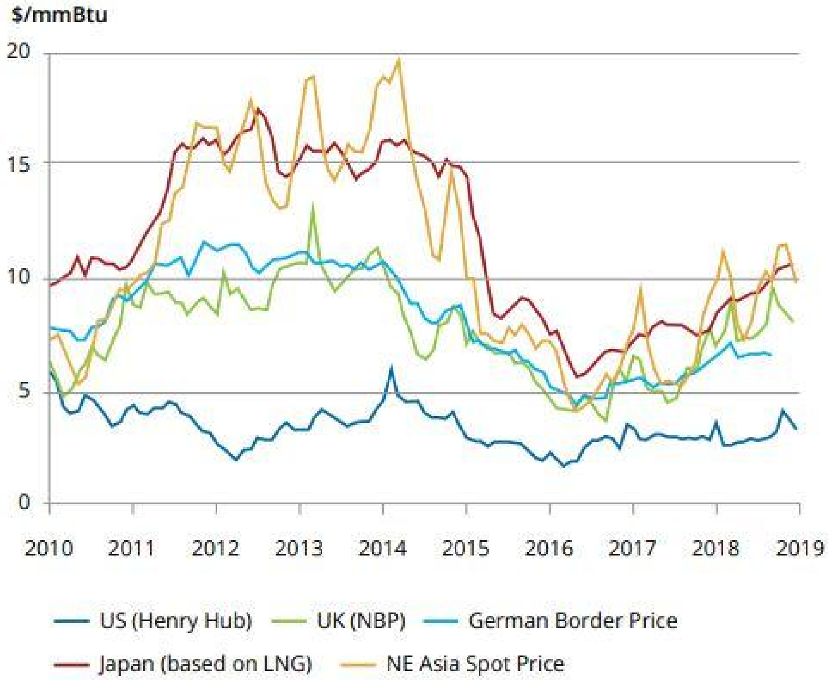


Fig. 2.2 Average Regional Natural Gas Prices [8]

Currently, HFO still remains one of the cheapest marine fuels available to burn outside the ECAs. Inside of them, however, LNG and LPG stand as the cheapest fuels to utilize. The major oil and gas company, Shell, has been offering to ensure an LNG price at least 20% lower than the MGO price for about 8 years in an effort to support lower carbon content fuels [9].

For the third consecutive year, LNG continues to set records in global trade. In 2014, 241.1 million tons (MT) was traded and 244.8 MT in 2015. In 2016, 258 MT of LNG was traded globally marking an increase of more than 5% (13.1 MT) in just one year. A growth rate of about 0.5% over the past four years, without any significant LNG supplier additions, was also a very noticeable trend according to the International Gas Union's 2017 World LNG Report. With a spot price of only $5.52/MMBtu in 2016, LNG accounted for approximately 25% of global energy demand, of which 9.8% was supplied as LNG [10].

Fig. 2.3 shows LNG trade volumes from 1990 to 2016. These include global regasification capacity, the total volume of LNG trade, and the number of exporting and importing countries.

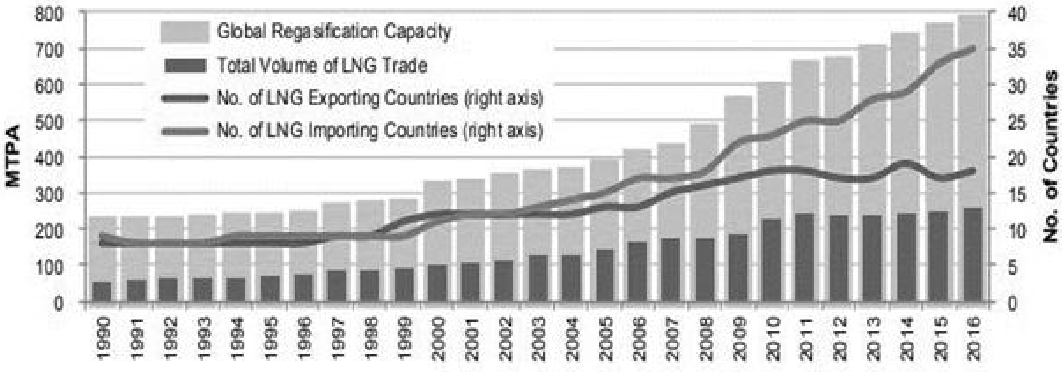


Fig. 2.3 LNG Trade Volumes [10]

Availability as a factor for being a successful alternative fuel must include the appropriate infrastructure in order to meet the growing levels of global demand. Meeting the demand challenges of an emerging LNG market means increasing the global regasification capacity, building new liquefaction plants, increasing the LNG shipping fleet, and increasing the number of proposed projects on natural gas discoveries. All of these factors play huge roles in the LNG market, ultimately deciding whether or not it is a viable resource or not. Fortunately, all of the above-mentioned factors are either increasing in number or currently satisfy global demand meaning that LNG as an alternative fuel is unquestionably an economically viable one [11].

Global regasification capacity increased from 776.8 million tons per annum (MTPA) to 794.6 MTPA in 2017 in just one year with more regasification terminals set to be complete in an array of countries. Global liquefaction capacity grew from 305 MTPA to 340 MTPA, from 2016 to 2017, respectively. New liquefaction proposals took a slight dip from a capacity of 890 MTPA to 879 MTPA from 2016 to 2017, respectively. This is because there has been an abundant number of gas field discoveries both globally and due to the US shale revolution affecting market demand. Because supply additions outpaced demand growth in 2016, many projects did not go forward as planned, however, because supply is high, LNG spot prices dropped by about $2 making it the cheapest marine fuel available [10] [12].

2.4 Emissions of Dual-Fuel Engines

The last reason that LNG makes a respectable alternative fuel to traditional MDO is its capability to meet the IMO emissions regulatory requirements. MAN B&W ME/ME-C-TⅡ, the same class of the main propulsion engine on the MV Ilshin *Green Iris*, engine performance parameters comply with Tier Ⅱ emission regulations because LNG contains a small sulfur content. The reasons this engine was selected for the MV Ilshin *Green Iris* are many, however, complying with strict Tier Ⅱ NOx limits and the 2020 sulfur cap were influential motives.

The ME-GI dual-fuel low-speed engine can operate on either HFO or LNG. When comparing the emissions from HFO and LNG the difference in SOx, NOx, PM, and CO_2 are substantial. The shift from HFO to LNG results in reductions (by percent difference of g/kWh) by approximately 92%, 26%, 37%, and 23%, respectively. This comparison is shown in **Table 2.3** comparing an HFO burning 6S70ME-C engine to a gas burning 6S70ME-GI engine both operating at 100% load. Due to LNG having a significantly low carbon content, some GHG emission contribution reports claim a CO_2 reduction of about 20% relative to MDO [13].

Table 2.3 Emissions Comparison from HFO and Gas Burning Engines [13]

Estimated Emissions 6S70ME-C		Estimated Emissions 6S70ME-GI	
Load 100%	**g/kWh**	**Load 100%**	**g/kWh**
CO_2	577	CO_2	446
O_2 (%)	1359	O_2 (%)	1340
CO	0.64	CO	0.79
NO_x	11.58	NO_x	8.76
HC	0.19	HC	0.39
SO_x	10.96	SO_x	0.88
PM (mg/m^3)	0.54	PM (mg/m^3)	0.34

Methane slip is a loss of unburned methane and is categorized into two types: operational emissions and engine emissions. Operational methane slip, like during bunkering operations, includes minor methane amounts having to be vented into the atmosphere. Methane slip is generally higher at lower engine loads. However, the ME-GI engine design boasts a 0.2% slip at low loads and a negligible slip at loads higher than 15% [14].

The Energy Efficiency Design Index (EEDI), mandated by the MEPC, regulates the grams of CO_2 per transport work as a function of installed power, specific fuel consumption, DWT, and speed. Marine propulsion engines typically have an NCR about 10% lower than MCR. This margin for the MV Ilshin *Green Iris*, however, is about 23% due to the EEDI and minimum propulsion power requirements established by the MEPC and the Maritime Safety Committee.

2.5 Fuels and their Properties

Having two types of fuel to utilize, HFO and LNG, the ME-GI engine onboard the MV Ilshin *Green Iris* must have standard fuel specifications to follow in order to achieve consistency in engine performance. **Table 2.4** shows the guiding specifications of the HFO to be used as a standard for the ME-GI engine.

Table 2.4 ME-GI Engine Guiding Specifications for HFO

Specification	**Value**	**Unit**
Density at 15℃	≤1.010	kg/m^3
Kinematic viscosity at 100℃	≤55	cSt
Kinematic viscosity at 50℃	≤700	cSt
Flashpoint	≥60	℃
Pour point	≤30	℃
Carbon residue	≤20	% (m/m)
Ash	≤0.15	% (m/m)
Water	≤0.5	% (v/v)
Sulfur	≤0.45	% (m/m)
Vanadium	≤450	mg/kg
Aluminum + Silicon	≤60	mg/kg

High engine performance for the ME-GI engine is still possible even if the lower calorific value (LCV) of the pilot fuel is about 38MJ/kg. Anything below this specific energy will require a pilot fuel amount above 3% for this engine.

Natural gas (NG) contains mostly CH_4 and higher hydrocarbons like ethane, propane, and butane (C_2H_6, C_3H_8, C_4H_{10}). ME-GI engines are capable of operating on a wide range of gas compositions, however, for better engine performance and specific gas consumption (SGC) the ME-GI engine operates by design on gas with an LCV of 50MJ/kg. **Table 2.5** shows the guiding specifications for fuel gas (LNG) to be used as a standard for the ME-GI engine [15].

Table 2.5 ME-GI Engine Guiding Specifications for LNG

Specification	**Value**	**Unit**
Lower calorific value (LCV)	≥38	MJ/kg
Methane (CH_4)	≥82	% (mol)
Ethane (C_2H_6)	≤15	% (mol)
C_3H_8 + C_4H_{10}	≥5	% (mol)
Higher order hydrocarbons (C_5H_{12} and higher)	≤1	% (mol)
Hydrogen sulphide(H_2S) + carbonyl sulphide(COS)	≤5	mg/M㎥
Nitrogen (N_2)	≤15	% (mol)

LNG must be cooled down to -162℃, and because of this, its hydrocarbon mixture composition has quite narrow limitations. Impurities such as water (H_2O), mercury (Hg), ammonia (NH_3), and carbon dioxide (CO_2) have already been removed, as well as heavy hydrocarbons (C_{2+}), as much as possible prior to liquefaction and bunkering. This is accomplished by a gas treatment system involving mercury removal units, acid gas removal units, dehydration systems, and heavy hydrocarbon removal systems. Most bunkered LNG contains, on average, anywhere between 80-90% CH_4. This means that the ME-GI engine can operate on less-than-average LNG composition (C_2H_6 = 15%) making it a flexible engine. This might, however, affect the fuel's ignition quality. When LNG is used as a fuel source, variations in energy content and fuel density may be encountered due to the relative amounts of methane and higher order hydrocarbons such as ethane and propane.

Cetane number (CN), methane number (MN), energy content, density, lubricity, and viscosity are all fuel qualities that significantly contribute to engine performance. Maintaining engine speed and load is crucial for smooth mode transition from diesel to gas mode. If a highly varied composition of gas fuel is supplied to the engine, it directly affects the amount of injected pilot fuel. Therefore, fuel qualities like CN (an indicator of combustion speed and compression needed for ignition) and MN (an indicator of ignition resistance), in a significantly variable supplied gas fuel, need to be accurately monitored for reliable engine performance and behavior.

The LNG that is supplied to a ship's LNG tank changes composition over time. This process is called aging and is due to the unavoidable heat-influx from the tank's surroundings. This heat-influx causes vaporization of lighter compounds, CH_4 and N_2, and the gas produced is referred to as boil-off gas (BOG). Consequently, the bunkered LNG will not have the same qualities by the time it is delivered to the engine. If the nitrogen content of the bunkered LNG is higher than 15% (mol), then it can be dealt with by decreasing engine load or increasing the amount of injected pilot fuel in order to maintain ignition qualities.

LNG has a much higher heating value (HHV) compared to HFO. That is about 55.2 MJ/kg and 41.8 MJ/kg, respectively. The lower heating value (LHV) of LNG and HFO is 48.6 MJ/kg and 39.0 MJ/kg, respectively. Fuel composition varies significantly across the globe, and its quality may affect the heating value by a range of about 5-10%. The auto-ignition temperature for diesel is about 245℃ whereas NG is about 704℃. This means NG requires a means of ignition via pilot fuel oil. Natural gas, however, has a high octane rating proving it suitable for engines with high compression ratios. This makes it a prime candidate among alternative fuel options for engines operating under the diesel cycle [16].

2.6 Combustion

Theoretically, ideal combustion can be achieved if four conditions of fuel injection that occur inside the engine cylinder are met: atomization, penetration, distribution, and dispersion. In diesel mode, HFO is the only fuel that is injected into the cylinder. In gas mode, both vaporized LNG and the pilot fuel oil (HFO) is injected into the cylinder. Assuming the guidance specifications for the fuel have been met (fuel qualities and proper injection temperature and pressure) there should be proper atomization of the fuel from the fuel injector and depth of which the fuel has penetrated into the cylinder. The ideal air-fuel ratio (AFR) varies with fuel composition, however, if met, ideal combustion can take place if the injected fuel was also dispersed and distributed throughout the combustion space. Ideal flame propagation will be achieved if all of these criteria have been met then forcing the piston down as combustion of the fuel/air/gas produces a flue (exhaust) gas with a higher density than the original mixture. A detailed explanation of the 2-stroke diesel cycle is shown in **Fig. 2.4**.

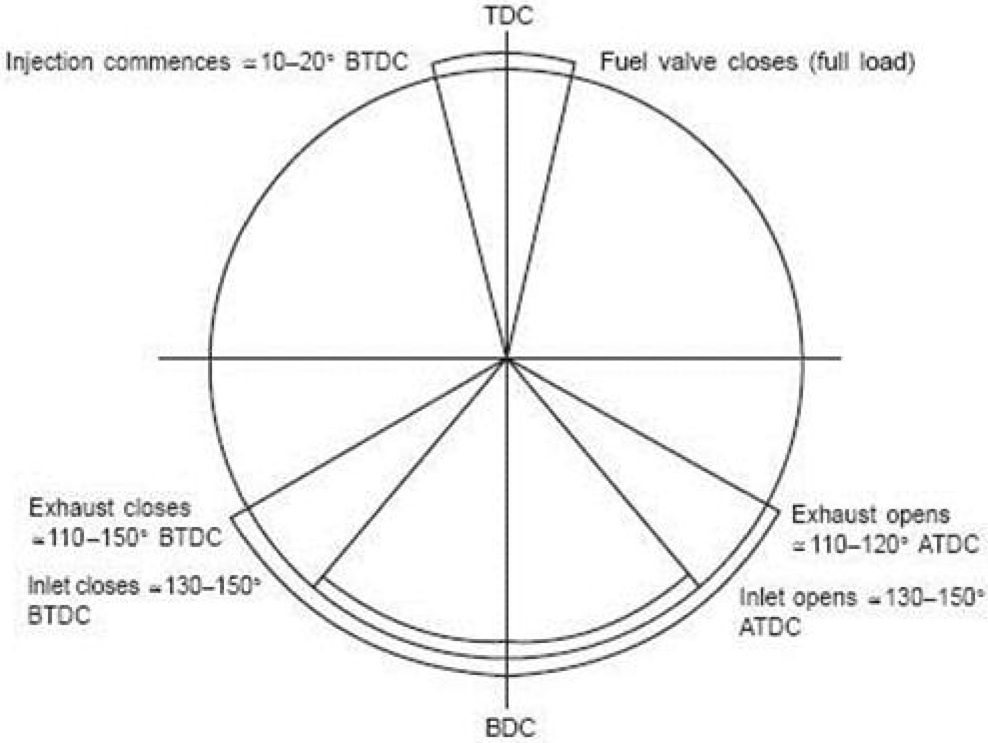


Fig. 2.4 Example of a 2-Stroke Engine Timing Diagram [17]

Diesel engine designs can vary greatly, and the crank angle per each event can differ. The ME-GI engine onboard the MV Ilshin *Green Iris* contains exhaust valves with uniflow scavenging. **Fig. 2.4** shows the event timing similar to said engine. Due to variable valve timing which helps engine performance and efficiency, modern engines typically do not have permanent event timing angles.

The cycle of **Fig. 2.4** is described as follows: After the power stroke, the piston rotates towards BDC. From 110 to 120 ATDC (after top dead center) the exhaust valve opens and the exhaust starts to emit from the cylinder. Before BDC, from 130 to 150 ATDC, the piston exposes the scavenging air ports at the bottom of the cylinder. At this time, both the exhaust valve and scavenging air ports are open causing scavenging to occur. The remaining exhaust gases are then forced out of the cylinder by the high pressure scavenging air. As the piston rotates from 130 to 150 BTDC the scavenging air ports close. Then from 110 to 150 BTDC, the exhaust valve closes and compression of the charged scavenging air begins. Near the end of the compression, from 10 to 20 BTDC, fuel is injected into the compressed air and ignition begins after proper fuel atomization, penetration, distribution, and dispersion. Upon ignition, expansion begins, the piston is pushed downwards, and the cycle repeats itself.

The combustion of fuel oil inside of a diesel engine cylinder typically occurs in four phases: the ignition delay period, rapid combustion period, steady burning period, and the afterburning period. These stages of combustion in a diesel cylinder can be seen in **Fig. 2.5**. The ignition delay period is the interval between when the injector opens and the start of ignition. There is usually no noticeable increase in cylinder pressure, until ignition occurs, had no injection occurred. The ignition delay period is mainly a function of the CN, or ignition quality, of the fuel and because LNG has a CN of almost zero, requires a pilot fuel for ignition to occur. Diesel fuels have a CN of about 44.

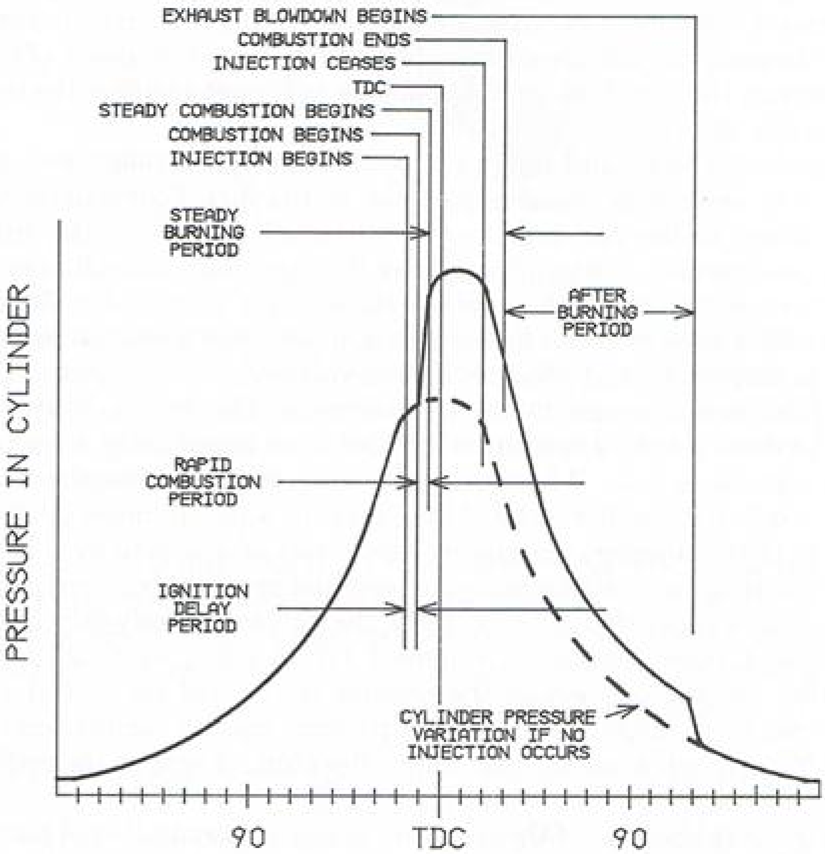


Fig. 2.5 Stages of Combustion in a Diesel Cylinder [18]

In the rapid combustion phase, the fuel has accumulated inside the cylinder and is accompanied by a sharp increase in cylinder pressure. During the steady burning period, the fuel that is entering the cylinder will burn immediately upon penetration, heating, vaporization, and mixing with charged air. Around the middle of this phase, the cylinder pressure will usually peak just after TDC and begins to fall just after injection cutoff.

If all the fuel in the cylinder has burned completely by the end of the steady burning period, the pressure will be smooth through the expansion stroke. Since irregularities normally exist due to incomplete combustion, however, the afterburning period produces SOx, NOx, PM, and other pollutants in what is known as chemical end reactions.

The chemical composition of HFO can vary greatly during the refinery process, and the carbon content of HFO can range from C12-30. Because of this, the following are possible and ideal combustion reactions for HFO:

Dodecane ($C_{12}H_{26}$)	$C_{12}H_{26} + 18.5O_2 \rightarrow 12CO_2 + 13H_2O$
Icosane ($C_{20}H_{42}$)	$C_{20}H_{42} + 30.5O_2 \rightarrow 20CO_2 + 21H_2O$
Pentacosane ($C_{25}H_{52}$)	$C_{25}H_{52} + 38O_2 \rightarrow 25CO_2 + 26H_2O$

The stoichiometric AFR for each of these HFO compounds is all about 15:1 with a variance of about 1%. This suggests that the chemical composition of the HFO does not inherently affect the variance in emissions produced by combustion but rather just the relative amount of CO_2 and H_2O produced.

In gas mode, the same exact process takes place, but just as the scavenging air ports are closed off by the piston as it rises from BDC, vaporized LNG is injected into the cylinder mixing with the scavenging air. The following is an ideal combustion reaction for LNG composed of pure methane:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

The stoichiometric AFR for LNG is about 17.3:1. This means that when in gas mode the amount of oxygen required for ideal combustion will be higher compared to when operating in diesel mode. Yet, the carbon content of methane is inherently much lower resulting in significantly reduced CO_2 emissions. **Table 2.6** shows more saturated hydrocarbon combustion reactions and their theoretical AFR.

Wartsila estimates its 32DF engine design will help lower its CO_2 emissions by as much as 93,000 tons per year. Because the ME-GI engine operates with pilot fuel with a higher carbon content than a pure gas-air mixture, in gas mode the annual CO_2 emissions will be higher than this claim [19].

Table 2.6 Saturated Hydrocarbon Combustion Reactions and the Theoretical Air-Fuel Ratio

Saturated Hydrocarbon (C_mH_{2m+2}) Theoretical Air-Fuel Ratio		
$C_xH_vO_z$ + (x+y/4-z/2)O_2 = xCO_2 + y/2H_2O		Air mass/Fuel mass (kgf)
CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	17.195
C_2H_6	$C_2H_6 + 3.5O_2 = CO_2 + 3H_2O$	16.024
C_3H_8	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	15.644
C_4H_{10}	$C_4H_{10} + 6.5O_2 = 4CO_2 + 5H_2O$	15.428
C_5H_{12}	$C_5H_{12} + 8O_2 = 5CO_2 + 6H_2O$	15.284
C_6H_{14}	$C_6H_{14} + 9.5O_2 = 6CO_2 + 7H_2O$	15.220
C_7H_{16}	$C_7H_{16} + 11O_2 = 7CO_2 + 8H_2O$	15.146
C_8H_{18}	$C_8H_{18} + 12.5O_2 = 8CO_2 + 9H_2O$	15.100
C_9H_{20}	$C_9H_{20} + 14O_2 = 9CO_2 + 10H_2O$	15.068
$C_{10}H_{22}$	$C_{10}H_{22} + 15.5O_2 = 10CO_2 + 11H_2O$	15.051

When characterizing combustion of a dual-fuel engine, heat release from the fuel that is injected into the engine is measured. Even though the ME-GI engine is a compression ignition (CI) engine, it shares both CI and spark ignition (SI) engine heat release characteristics. Traditionally, three to four parts are examined during heat release: combustion of the diesel pilot fuel, combustion of methane in the premixed pilot-region, flame propagation through the methane-air mixture, and bulk ignition of the end gas. **Fig. 2.6** shows the rate of heat release between diesel and LNG expressed as the total heat release in MW per crank angle in degrees ATDC at 75% engine load of a ME-GI engine.

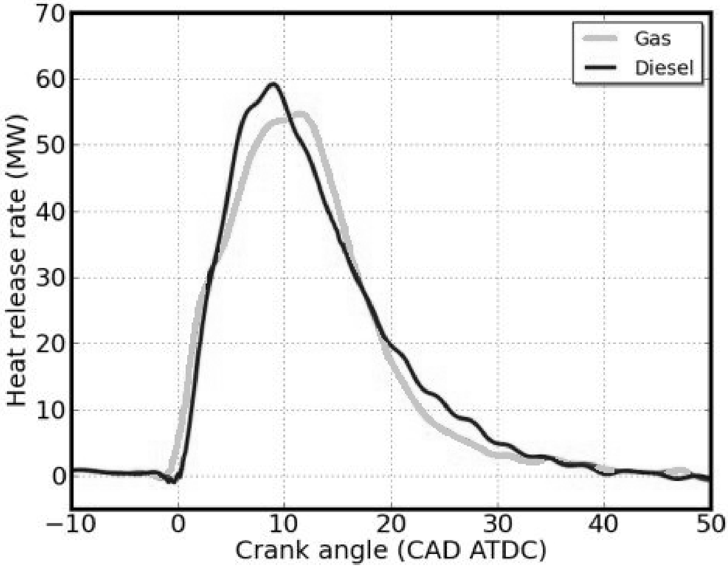


Fig. 2.6 Diesel and Gas Injection ROHR [20]

Due to each fuel having different properties, the combustion characterization of each fuel is different. LNG has a similar heat release profile to that of diesel, however, it can be seen that LNG has a slightly steeper ROHR just after ignition normally caused by the pilot fuel combustion. The total ROHR for LNG is also slightly smaller than diesel ATDC.

Substitution rate is a parameter that largely determines combustion characteristics while the engine is in dual-fuel mode. The substitution rate describes the amount of energy supplied by LNG expressed as a percentage of the total energy that would be provided by diesel. The percentages of LNG and diesel substitution against the engine load is shown in **Fig. 2.7.**

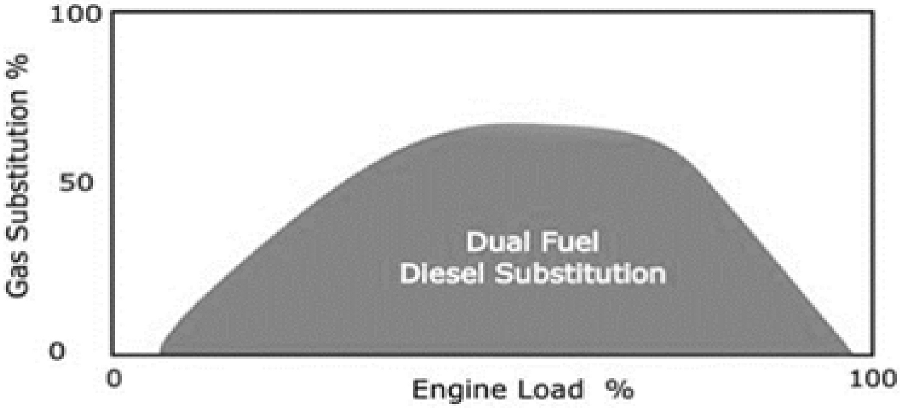


Fig. 2.7 Substitution Rate [21]

Various studies have found that there is an optimal engine load range at which gas substitution is maximized. If operating conditions allow (operating outside of an ECA), this substitution rate could be applied to optimize engine performance in SDF mode. Where gas substitution is maximized represents the optimal engine load range. At the high and low ends of the engine load range, gas substitution is 0%. These findings show that dual-fuel engine substitution is significantly affected by engine load and speed, which in turn, affects the engine performance. Comparisons with SDF mode, however, will not be discussed in this paper. During mode transition substitution rate is very critical for maintaining smooth engine operation and will be discussed [22].

2.7 Mode Transition

Transitioning from liquid fuel to gaseous fuel is one of the major advantages of the ME-GI or any dual-fuel engine. This key benefit, however, has some consequences. Despite its attraction to environmental concerns of utilizing a cleaner burning fuel, dual-fuel engines have had a slower acceptance than what the market and manufacturers predicted. This is because of the lack of an LNG bunkering infrastructure worldwide and also higher capital expenditures. Those issues aside, as they can be gradually addressed long-term, there are other disadvantages to utilizing dual-fuel engines [23].

Besides safety challenges, there are several technical challenges of operating a dual-fuel engine. These include sustaining operating limits and constraints, fuel variability, and control during mode transitions. Changing over from gas mode to diesel mode, as reported by several engine manufacturers, takes only about one second at any load. Switching back from diesel to gas mode again can take up to several minutes but can only be performed under specific load circumstances. In regards to this, one specific challenge is the combustion characterization during the dynamic fuel substitution of mode transition from diesel to gas mode. With limited data and access to testing facilities of dual-fuel, low-speed marine engines combustion characterization becomes very problematic and complicated [24].

Dual-fuel engines fulfill economic and environmental benefits through the combustion of different ratios of two types of fuel in different operating modes. A critical aspect while controlling transitions between operating modes is to maintain constant total fuel energy. There are several difficulties with today's applied methods used to do so. In some instances, at a selected gaseous fuel supply, the gas AFR may fall out of range. Even if the total fuel energy remains the same, this can result in the engine exhibiting power droops and surges. If excess air is provided to maintain the desired AFR, there are other parameters that affect combustion. Fuel variability and energy content can contribute significant error to the control devices, used during mode transition, which is based on fuel energy content and AFR.

One method of control during mode transition is by using a kind of combustion index based on operating conditions. The combustion index then provides desired engine operation at multiple gas and liquid fuel ratios. This technology can be utilized while transitioning between modes and also during gas mode to ensure adequate pilot fuel amount and gas injection control [25].

Chapter 3: Data Analysis and Results

3.1 Test Ship and Research Methodology

Comparative research typically has no peculiar methodology. The data that was collected, however, can be quantified, verified, and is amenable to statistical manipulation. Therefore, quantitative analysis was deemed suitable.

Data for this study was collected directly onboard the MV Ilshin *Green Iris*, as shown in **Fig. 3.1**, by two colleagues, researchers from the Green Energy Center at the Korea Maritime and Ocean University, as well as a software engineer. The software, provided by NAPA, being utilized by the MV Ilshin *Green Iris* served as a means to collect the data presented in this paper. My colleagues boarded and disembarked the vessel in Gwangyang, South Korea on April 9th, 2018 and were able to collect specific engine and navigational data for a period of about 24 hours. **Fig. 3.2** shows its current domestic route from Donghae to Gwangyang, South Korea.

Fig. 3.1 MV Ilshin *Green Iris* [26]

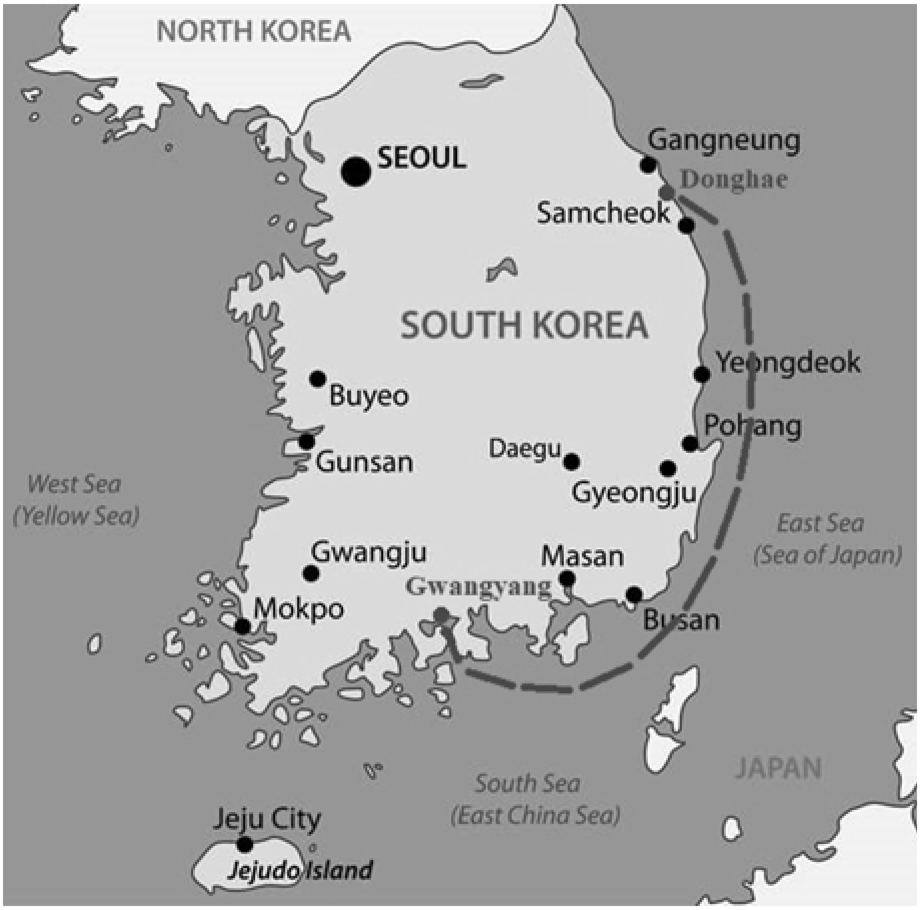


Fig. 3.2 Voyage Route of the MV Ilshin *Green Iris*

Navigational data such as water depth, wind speed and direction, heading, rudder angle, geographical coordinates, and longitudinal water speed were collected. These factors, however, were not considered while examining the engine data and analyzing engine performance. The engine data alone is what concludes whether or not proper engine performance and efficiency is achieved. Parameters such as ship speed can be a determiner of propeller efficiency, however, require more data than what was collected in order to calculate. Other parameters such as hull condition, weather/sea state, etc., would have needed to be collected in order for navigational data to be effectively used. The engine data collected is comprised of the following: dual-fuel state of the engine, RPM, power relative to MCR, fuel oil load, gas fuel load, gas flow, gas inlet pressure and temperature, and fuel oil LCV, sulfur content, temperature and density, and pressure mean indicator (PMI) data. In addition to the data collected onboard, sea trial results of the MV Ilshin *Green Iris* were released by the shipyard that conducted the tests in January of 2017. Shop test data provided by the Hyundai Mipo Dockyard will also be used in order to compare between modes. **Table 3.1** shows the MV Ilshin *Green Iris*' main engine specifications.

Table 3.1 MV Ilshin *Green Iris* Main Engine Specifications

Type:	One, Hyundai-MAN B&W 6G50ME-C9.5-GI-TⅡ
Nominal Rating:	10,320 kW x 100.0 rpm
Maximum Continuous Rating (100%)	7,250 kW x 88.7 rpm
Nominal Continuous Rating (77.2%)	5,597 kW x 81.4 rpm SM 10% 5,088 kW x 78.8 rpm ×14 kts
Cylinder No. x Bore x Stroke	6 cylinder x 500 mm x 2,500 mm
The main engine type designation is as follows: 6 - Number of cylinders G - Green, Ultra-long stroke 50 - Diameter of piston (cm) E - Electronically controlled C - Compact engine	9 - Mark number .5 - Version number GI - Gas injection TII - IMO Tier Level

It should be noted that a typical engine margin between NCR and MCR is usually about 10%. As per established IMO regulations, a bulk carrier with a DWT of less than 145,000, like the MV Ilshin *Green Iris*, must increase its NCR relative to MCR margin in order to maintain the maneuverability of the ship in adverse weather conditions meeting minimum propulsion power requirements. Therefore, following appropriate calculations and EEDI regulations, the vessel was built with a resulting engine margin of approximately 23%.

For the duration the data was being collected onboard, the vessel did not undergo any bunkering operations for HFO or LNG. The HFO inside the settling and service tanks, being from the same source (meaning same quality), therefore, had an unchanging HFO sulfur content (3.2%, which is higher than the ≤0.45% guiding specification shown in **Table 2.4**) and LCV of 42.41MJ/kg. Furthermore, the HFO was heated consistently and delivered to the engine at 116℃. The HFO in the settling tanks was heated to approximately 60℃ and kept at a consistent density of 990.3 kg/m^3 at 15℃ resulting in a kinematic viscosity of approximately 180 mm^2/s at 50℃. The fuel was delivered to the engine at about 116℃ with a viscosity of approximately 12 cSt. All of these parameters meet the guiding specifications for HFO as shown in **Table 2.4** in section 2.5.

Due to strict IGF code, LNG is sold based on its energy content which is dependent on its actual composition and temperature. The LNG quality for the ME-GI engine must have an LCV guiding specification of 38 MJ/kg and must be purchased and bunkered as such. LNG quality data was not collected during the time onboard, but for this reason, it can be safely assumed that the LNG quality met the guiding specifications as shown in **Table 2.5** in section 2.5. Because the fuel being used during data collection had a quality that meets the engine design criteria, we can eliminate most errors, if any, that could be caused by fuel quality.

3.2 Power Curves

Power curves were created by plotting engine power against engine RPM. Two power curves for when the engine was operating in diesel and gas mode were plotted. **Fig. 3.3** shows the power curve when the engine is in diesel mode.

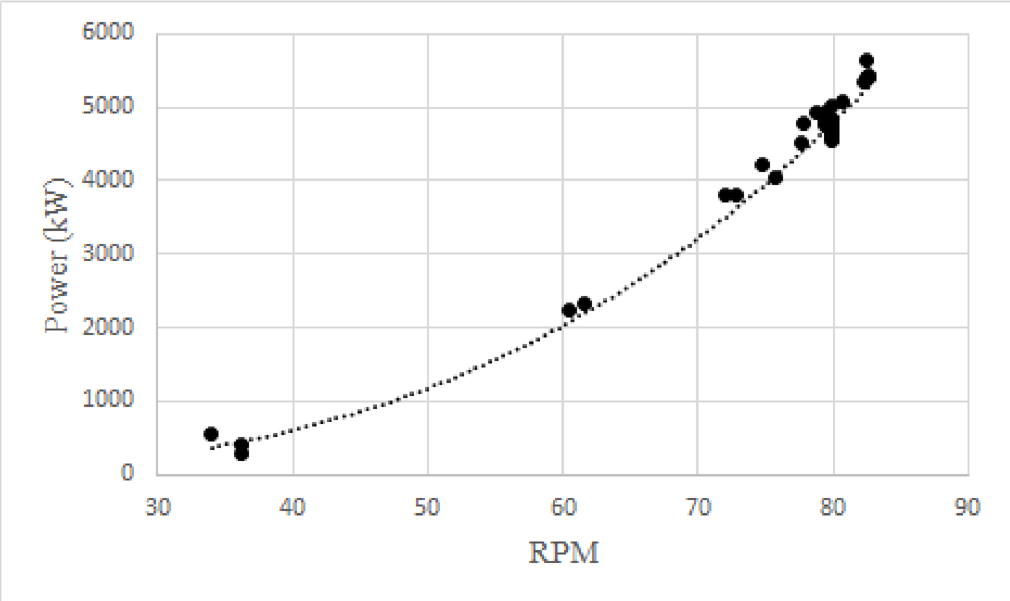


Fig. 3.3 Diesel Mode Power Curve

A trendline was added to the graph in **Fig. 3.3** representing the propeller curve. As RPM increases, the power output increases but the rate at which it increases decays. This is due to the inherent design of the engine, in particular, its volumetric efficiency. Generating more power requires more fuel, in turn, requiring more air to complete the combustion process. At the point of maximum power, the engine's volumetric efficiency is also at its highest meaning that the engine is drawing in the maximum mass/volume of air mechanically possible. Because this is a well-known principle of operation, the engine is rarely operated at an RPM that results in higher than peak power output. Therefore, the data collected contains no engine performance figures beyond peak power or MCR.

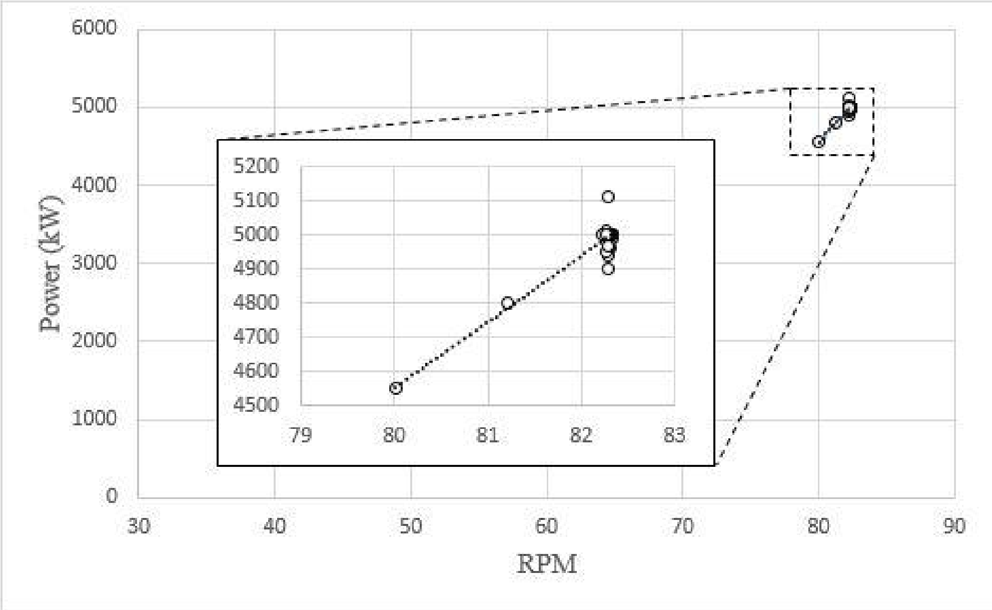


Fig. 3.4 Gas Mode Power Curve

Fig. 3.4 shows the power curve when the engine is in gas mode. There are operating limitations when the engine is in gas mode. In low load conditions, especially below 10%, the engine will typically be operating in diesel mode. This is because at low engine loads the amount of hydrocarbon emissions, or unburned fuel, increases. Because of the stoichiometric AFR inherent to LNG, there is an inability to provide an equivalent gas ratio at low engine loads. Poor flame propagation would exist resulting in exhaust gas temperatures too low to ensure acceptable emission percentages. For this reason, gas mode is typically enabled when the engine load is well above 15% to ensure proper exhaust gas temperatures have been reached before transitioning to gas mode. Therefore, the collected engine data while in gas mode was limited to higher engine speeds and loads. This data can be seen in the expanded graph inside **Fig. 3.4** with a trendline added representing the propeller curve.

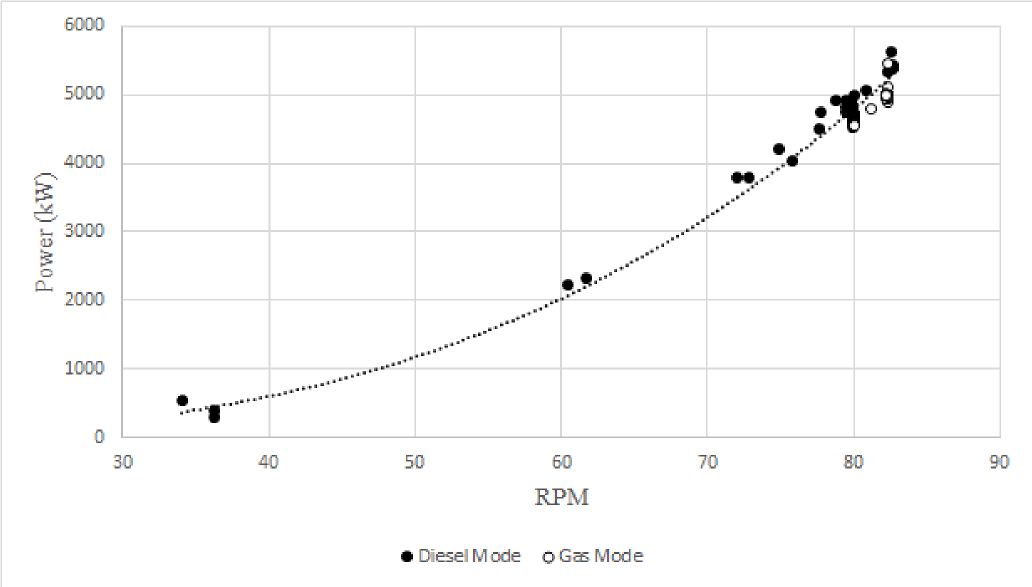


Fig. 3.5 Diesel and Gas Mode Power Curve

There are six operational dual-fuel states of the engine. Each of the states is thusly and correlate to if the dual-fuel capability is not ready, ready, starting, running, stopping, or stopped. The data in **Fig. 3.5** was collected sequentially while the dual-fuel state of the engine changed and underwent mode transitions from diesel mode to gas mode and then back to diesel mode. In order to validate dual-fuel operation, it must be proven that transitioning between modes is both safe and effective. To show that engine speed and power is not compromised during mode transition, the data during mode transition plotted in **Fig. 3.5** is shown more closely in **Table 3.2**.

Table 3.2 Mode Transition Data

Engine Load	Dual-Fuel State*	RPM	Power (kW)	Fuel Oil Load (%)	Gas Fuel Load (%)
62.5	2	79.93	4,550	62	0
63.6	3	79.94	4,580	64	0
63.5	3	79.92	4,600	63	1
66.4	4	80.01	4,550	6	61
66.5	4	81.22	4,800	5	62
…	4	…	…	…	…
76.0	4	82.30	5,450	5	71
69.4	4	82.28	5,000	5	65
70.8	1	82.23	5,080	75	0
71.3	1	82.29	5,200	71	0
*(1 = Not Ready, 2 = Ready, 3 = Starting, 4 = Running, 5 = Stopping, 6 = Stopped)					

It should be noted that as the engine went from diesel mode to gas mode and then back to diesel mode, this data was taken consecutively. The break in data represents about five hours of the engine operating continuously in gas mode and is not shown. **Table 3.2** only shows the engine data relevant during mode transition. As can be seen, transitioning between modes has a minimal effect on engine load relative to MCR as well as engine RPM and power output. This data proves considerably that LNG as an alternative fuel is as operationally effective as traditional HFO.

In addition to verifying engine performance stability during mode transition, it is also essential to compare engine performance of diesel mode and gas mode at specific engine speeds. This comparison is shown in **Table 3.3.**

Table 3.3 Mode Power Comparison by RPM

Diesel Mode		Gas Mode
Power (kW)	**RPM**	**Power (kW)**
5,080	82.23	5,000
5,000	82.24	5,000
5,100	82.26	5,010
5,000	82.27	5,000
5,100	82.28	5,000
5,100	82.29	5,110

At the same engine speeds, there is a noticeable difference in power between modes with an average difference of about 0.85%. This difference, expressed as a difference deducted from MCR, would mean a loss of approximately 61.68 kW of power when in gas mode. Given that when operating in gas mode significantly reduces engine emissions this is not a significant loss yet but a small sacrifice to make to meet IMO emission regulatory requirements. Bearing in mind that weather conditions, sea state, and various other navigational parameters are not taken into consideration, this loss may even be considered negligible.

3.3 Mean Effective Pressure

One of the best ways to directly compare between modes is by examining the mean effective pressure (MEP) from in-cylinder pressure over the complete engine cycle. From the data collected onboard the MV Ilshin *Green Iris*, the indicated mean effective pressure (IMEP) and the brake mean effective pressure (BMEP) was calculated.

In order to effectively compare between gas mode and diesel mode, calculations must be executed for both modes at the NCR: 5,597 kW x 81.4 rpm. Since NCR is measured to be at 81.4 rpm the closest measurement to this in gas mode was 4,800 kW at 81.22 rpm, and in diesel mode was 5,100 kW at 82.21 rpm. This is a percent difference of 0.22% and 0.99%, respectively, and will be taken into account in the final calculation. All power values were converted from kilowatts to newton-meters as well as all pressures from pascal to bars.

Firstly, the surface area of the piston, as well as the volume displacement of the cylinder, must be calculated:

$$S_p = \frac{\pi B^2}{4} = 0.196 \text{ m}^2$$

$$V_d = S_p L = 0.491 \text{ m}^3$$

where Sp is the surface area of the piston, B is the cylinder bore, Vd is the volume displacement of the cylinder, and L is the stroke length. The values needed to make these calculations can be found in **Table 3.1**. The Sp and swept volume, however, are estimated values because the real surface area of the piston has a highly complicated geometry and is guarded information not released to the public by the engine's manufacturer.

The BMEP was calculated from the dynamometer power (torque) and is the actual output of the engine at the crankshaft and does take into account the engine efficiency. BMEP for gas mode and diesel mode is calculated as follows:

$$BMEP_G = \frac{2\pi n_r T_e}{n_c V_d} = 12.04 \text{ bar}$$

$$BMEP_D = \frac{2\pi n_r T_e}{n_c V_d} = 12.64 \text{ bar}$$

where n_r is the number of crankshaft rotations for a complete engine cycle (1), T_e is the brake torque (at NCR), and n_c is the number of cylinders (6).

In order to calculate IMEP, indicated torque data is required but was not collected onboard the vessel. We can, however, make an assumption based on the following equation:

$$p_i = p_e + k_1$$

where p_i is the indicated engine power, p_e is the mean effective pressure, and k_1 is the mean friction loss which has been generally found to be approximately 1bar [27].

Through the previous equation, we can, therefore, make a theoretical calculation of IMEP. The result of this calculation can be calculated as follows for both gas mode and diesel mode:

$$IMEP_G = \frac{2\pi n_r T_i}{n_c V_d} = 12.04 + 1 = 13.04 \text{ bar}$$

$$IMEP_D = \frac{2\pi n_r T_i}{n_c V_d} = 12.64 + 1 = 13.64 \text{ bar}$$

where T_i is the indicated torque (at NCR) and other factors as previously stated. Because swept volume and the surface area of the piston are estimated values, and the assumption of the indicated engine power equation, the true value of IMEP cannot be calculated thus the resulting values are estimations. However, operation data from the vessel dated December 2nd, 2017, shows an average p_i across all six cylinders at a value of 14.0 bar in diesel mode. This is a difference of only about 2.60%.

Due to friction in the thrust bearing, the shaft power is approximately 1% less than the effective engine power. The dynamometer used to obtain the torque measurements used to calculate BMEP is located after the thrust bearing, however, this 1% loss is not considered in the calculation.

3.4 Thermal Efficiency

The thermal efficiency of an internal combustion engine can be defined as the fraction of heat that becomes useful work. By this definition thermal efficiency can be calculated with the following equation:

$$\eta_t = \frac{Output}{SFOC}$$

where η_t is the thermal efficiency, Output is the engine power output, or brake horsepower (BHP), and SFOC is the specific fuel oil consumption. Through this formula, after converting both the known engine output and SFOC (at a specific engine load) to kcal/h, it is possible to calculate the thermal efficiency.

The data required to perform thermal efficiency calculations at various engine loads was provided by Hyundai Heavy Industries' Hyundai Mipo Dockyard. The dockyard provided the official shop test results of the 6G50ME-C9.5-GI engine used onboard the MV Ilshin *Green Iris*. Among this data included the BHP and SFOC rates at 25%, 50%, 75%, 77.2% (NCR), 100% (MCR), and 110% engine load. Assuming one (1) calorie* is equal to 4.1846 J, the BHP and SFOC rates were converted to kcal/h with a fuel LCV of 10,200 kcal/kg in order to correct to ISO conditions. The calculation was performed for both diesel and gas mode across previously mentioned engine loads. In gas mode, both the pilot fuel and gas consumption rates were included in the total SFOC. The results of these calculations can be seen in **Fig. 3.6**.

* 1 calorie = 4.184 (thermochemical), 4.1868 (steam table), 4.186 (SI)

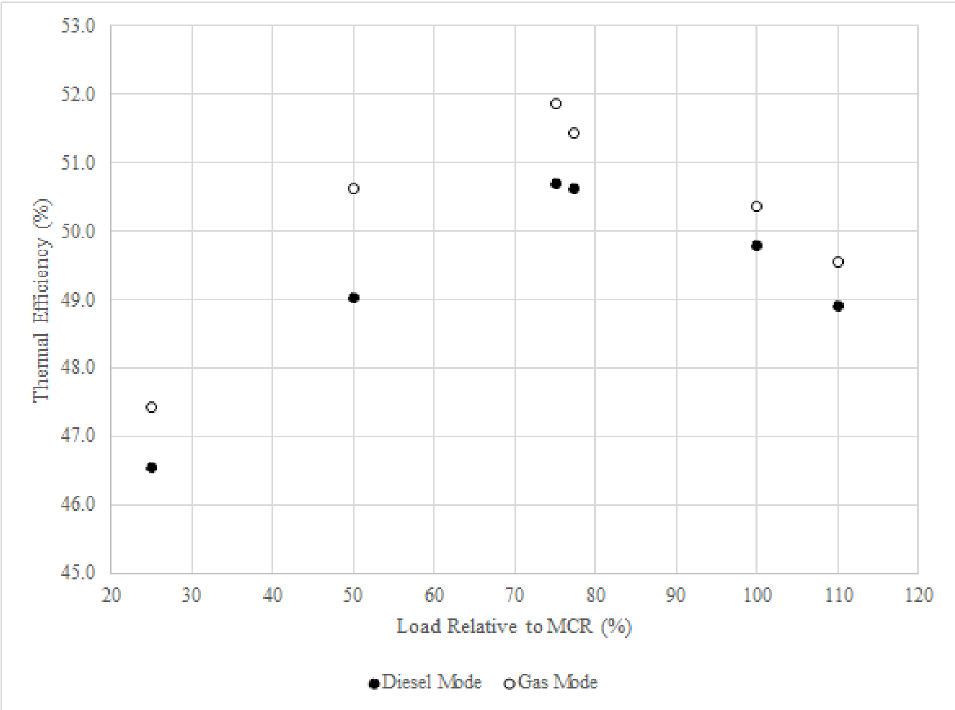


Fig. 3.6 Effect of Load on Thermal Efficiency by Mode

As can be seen in **Fig. 3.6**, gas mode has an overall higher thermal efficiency than diesel mode with an average percent difference of 1.91% in favor of gas mode. Diesel mode has a thermal efficiency of 50.6% and 49.8% at NCR and MCR, respectively. Gas mode has a thermal efficiency of 51.4% and 50.3% at NCR and MCR, respectively. Gas mode has an overall higher thermal efficiency because its SFOC is lower at each engine load. Gas mode uses less g/kWh of fuel because natural gas has a higher calorific value than diesel. According to the official shop test results, the Bunker-A fuel used during the testing period had an LCV of 10,021 kcal/kg and the gas used had an LCV of 49,455 kcal/kg.

The most significant difference between modes is the type of fuel used. A known issue of many dual-fuel engines is methane slip when operating in gas mode. Incomplete combustion of the LNG inside the combustion chamber can cause methane slip reducing engine efficiency. There are, however, more parameters that can cause efficiency loss.

The biggest difference between diesel and gas mode is the use of a different fuel for each mode. Put simply, fuel selection has impacts on efficiency. In general, higher fuel energy provides a higher potential work output. Fuels with simpler molecular formulas typically produce lower combustion irreversibility (% fuel energy). With LNG having a higher LCV and a simpler molecular formula than HFO, it would be intuitive to assume that operating a diesel engine with LNG would have higher overall efficiency. This is not always the case, however, because LNG has a CN of nearly zero, the addition of pilot fuel (HFO) and auxiliary equipment, such as exhaust scrubbers, EGR, SCR, etc., changes its natural combustion characterization causing a change in its overall efficiency.

3.5 Specific Fuel Oil Consumption

The specific fuel oil consumption (SFOC) rate is another effective way to compare between gas and diesel modes directly. **Fig. 3.7** shows the SFOC rate in g/kWh for each mode against engine load relative to MCR for the ME-GI engine onboard the MV Ilshin *Green Iris*.

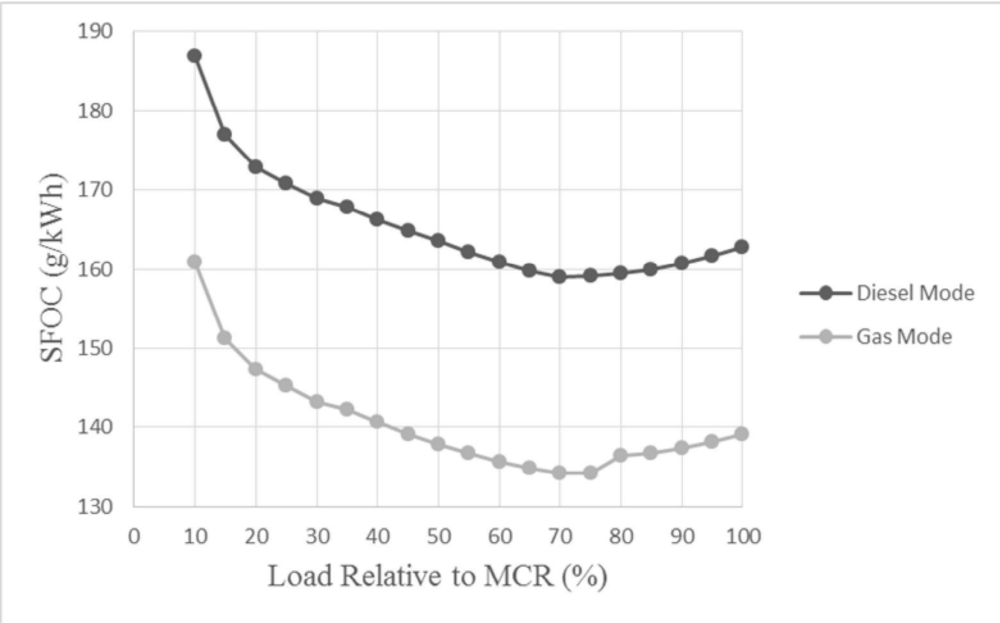


Fig. 3.7 Effect of Load on SFOC by Mode

The data in **Fig. 3.7** was calculated theoretically using a mathematical model provided by MAN Energy Solutions, formerly known as MAN Diesel & Turbo, through their CEAS (Computerized Engine Application System) Engine Calculations platform. These calculations assume ISO parameters with ambient air temperature and seawater temperature of 25℃. In gas mode, the SFOC includes the combined gas consumption rate and the pilot oil consumption rate. Between modes, the SFOC has an average percent difference of about 16.27%, favoring gas mode, yielding an SFOC average difference of about 24.88g/kWh. With a smaller SFOC rate than diesel mode, gas mode results in a more economical fuel choice than HFO. This is also largely due to the fact that LNG has a higher LCV than HFO in terms of mass.

3.6 Exhaust Gas

Exhaust gas temperature and amount are two parameters that can be compared to engine load to determine engine performance. **Fig. 3.8** shows the exhaust gas temperatures by mode against engine load relative to MCR. These exhaust gas values were measured before the turbocharger (post-cylinder) during shop tests at the Hyundai Mipo Dockyard.

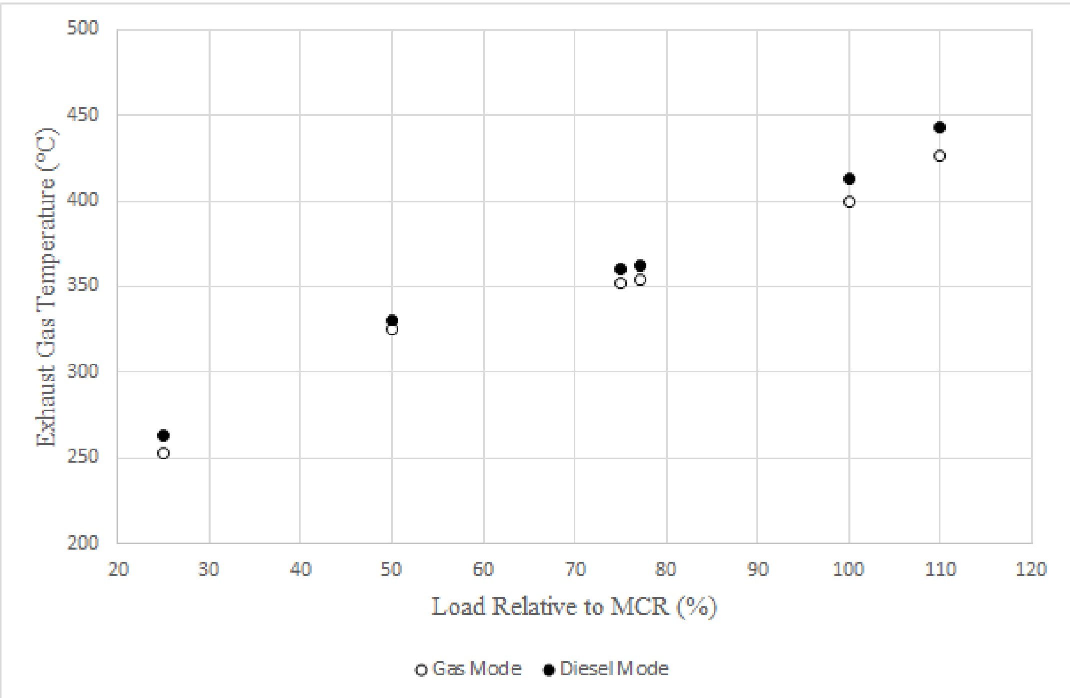


Fig. 3.8 Effect of Load on Exhaust Gas Temperature by Mode

In **Fig. 3.8** it can be seen that gas mode exhibits a lower exhaust gas temperature against engine load with an average difference of 10.17℃. The exhaust gas amount in kg/s was also calculated against engine load for each mode using the CEAS Engine Calculations platform (assuming ISO parameters as stated previously). These calculations conclude that gas mode operation yields less exhaust gas with an average difference of 0.032 kg/s. Over the course of a year, this means that operating in gas mode could result in up to approximately 996 metric tons in exhaust gas reductions.

It is normal to see an exhaust gas temperature increase of 50-60℃ from the shop test to the sea trial due to operation on HFO and altered climatic conditions. This is not the case for this engine, however, where post-cylinder exhaust temperatures actually decreased from the shop tests to the sea trials. The main reason behind this is because of outside air temperature differences when the shop test and sea trials were performed, during the summer (June) and winter (January), respectively [27].

In order to effectively compare exhaust gas temperatures, we must consider why there might be some deviations during actual engine operation. These reasons are shown in **Table 3.4**.

Table 3.4 Causes of Exhaust Gas Temperature Increase [27]

Cause	Max temperature increase (℃)	Notes
Turbocharger fouling (including air intake filter) and exhaust gas uptake	+30	Turbocharger exhaust gas back pressure: normal range: 100-300 ㎜WC Max back pressure at MCR: 350 ㎜WC
Air cooler fouling	+10	
Mechanical defect/deterioration	+10	
Climatic (ambient) conditions	+45	
Operation on heavy fuel, etc.	+15	
Total	110	

3.7 PMI Results

A pressure mean indicator (PMI) is a tool used for collecting engine cylinder pressure during engine operation. This allows cylinder pressure against °CA or relative cylinder volume to be plotted. Modern technology allows this data to be plotted on a chart in real time allowing for engine performance optimization. The MV Ilshin *Green Iris* utilized a particular PMI auto-tuning software developed by a highly acclaimed engine manufacturer to capture this data while fellow colleagues were onboard.

For this study, PMI data analysis is one of the best ways to compare between modes as it allows direct insight into engine event timing, performance, and combustion efficiency. PMI data was collected across various RPMs and engine loads, however, for a simpler analysis PMI data at NCR and MCR for both modes will be analyzed. **Fig. 3.9** shows the cylinder pressure against °CA in diesel mode and gas mode at NCR and has been averaged across all six cylinders.

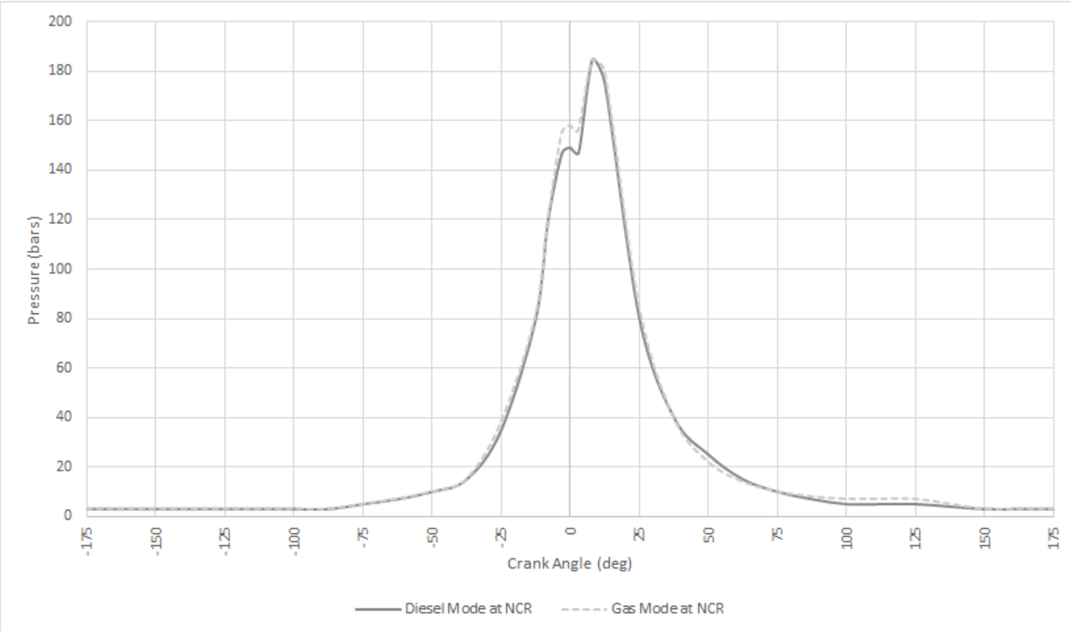


Fig. 3.9 Cylinder Pressure versus °CA for Diesel and Gas Mode at NCR

As can be seen from **Fig. 3.9**, at NCR there is little variation between diesel and gas mode except for approximately a 10 bar difference between modes in compression pressure. This difference will be addressed later. Across all six cylinders, the average maximum cylinder pressure for diesel and gas mode was 184.8 bars and 184.5 bars, respectively, with a 0.16% difference. Scavenging air pressure for diesel and gas mode at NCR was 2.11 bars and 2.03 bars, respectively, with a 3.86% difference. A slight drop in scavenging air pressure when in gas mode would be standard due to the fact that vaporized LNG has a higher stoichiometric AFR compared to diesel and requires more air per mass fuel (as discussed in section 2.6). This drop in scavenging air pressure, even though small, also validates that turbocharger efficiency has not been compromised because of the fuel type change.

Stages of combustion are described in Section 2.6 with reference to **Fig. 2.5**. It is very noticeable that the ignition delay period is apparently very short in gas mode at NCR. This may be due to the fact that LNG has a CN of nearly zero so the pilot fuel that has been mixed with the vaporized LNG starts to combust and expand immediately upon injection resulting in almost no significant ignition delay. This ignition delay period, however, may be more or less significant at other engine RPMs and engine loads.

Fig. 3.10 shows the cylinder pressure against °CA in diesel mode and gas mode at MCR and has been averaged across all six cylinders. Similar to the NCR results, there is also little variation between diesel and gas mode. Compared to the NCR results, there is a slightly steeper cylinder pressure drop after fuel injection (just before TDC) when in diesel mode. Despite this steeper fuel injection pressure drop, the average maximum cylinder pressure for diesel and gas mode was 185.1 bars and 184.1 bars, respectively, with a 0.54% difference.

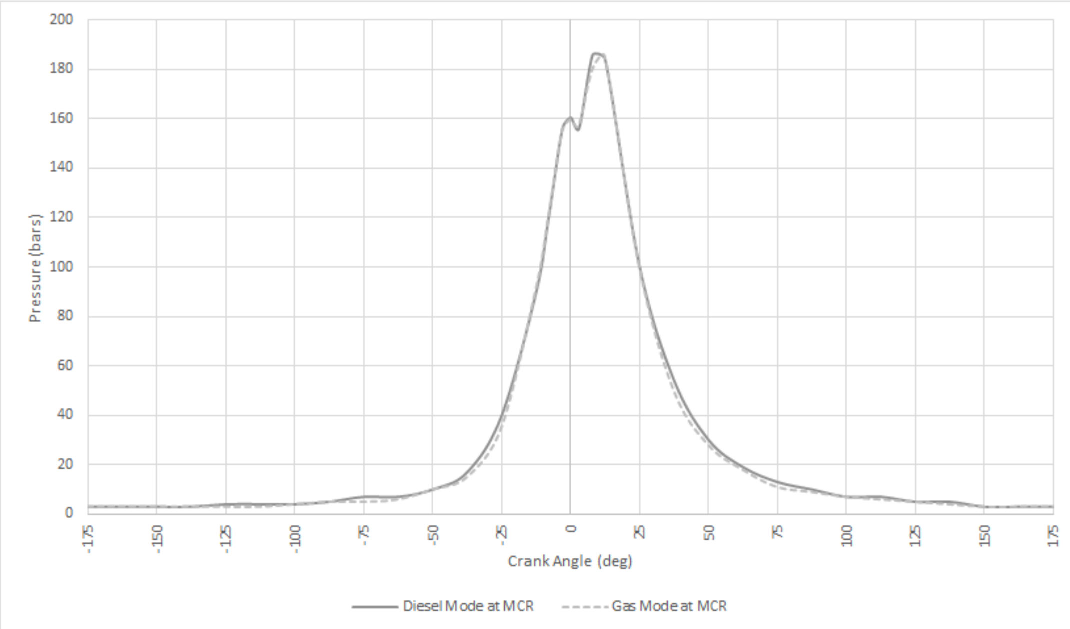


Fig. 3.10 Cylinder Pressure versus °CA for Diesel and Gas Mode at MCR

The combustion phases at MCR of diesel and gas mode are not significantly different in the traditional sense of combustion stage analyzation. At MCR, gas mode and diesel mode show similar Pcomp and Pmax pressures and a very similar ignition delay period length before rapid combustion begins. This may be due to the fact that vaporized LNG has a lower density than HFO at injection resulting in a relatively slightly higher cylinder pressure. The steady burning and after burning period in both modes at NCR and MCR have also very similar lengths.

Further results from the PMI measurements allow a comparison between the mean indicated pressure (Pi), compression pressure (Pcomp), maximum cylinder pressure (Pmax), and the scavenging air pressure (Pscav) of each mode at NCR and MCR. **Fig. 3.11** shows these comparisons as an average across all six cylinders.

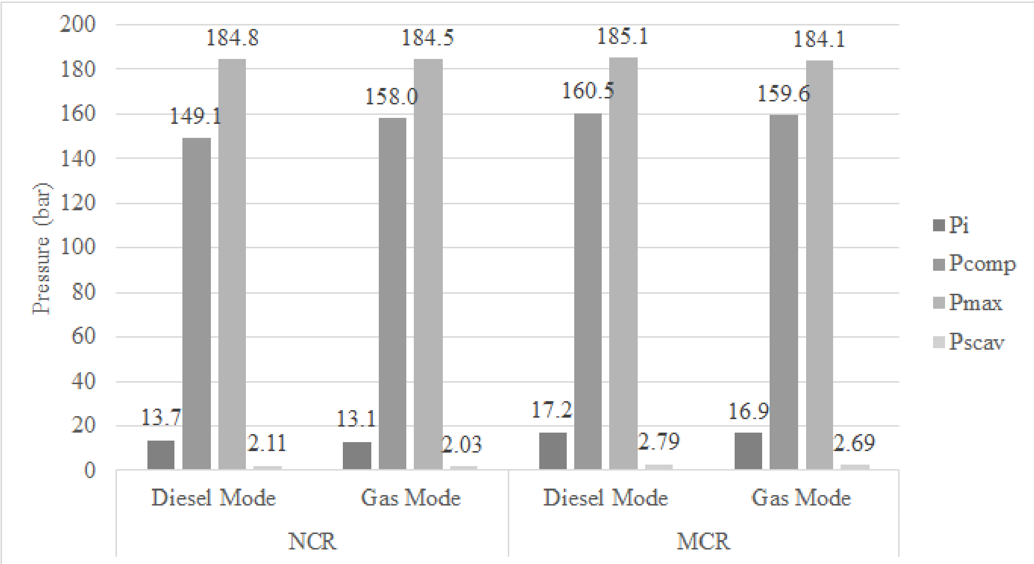
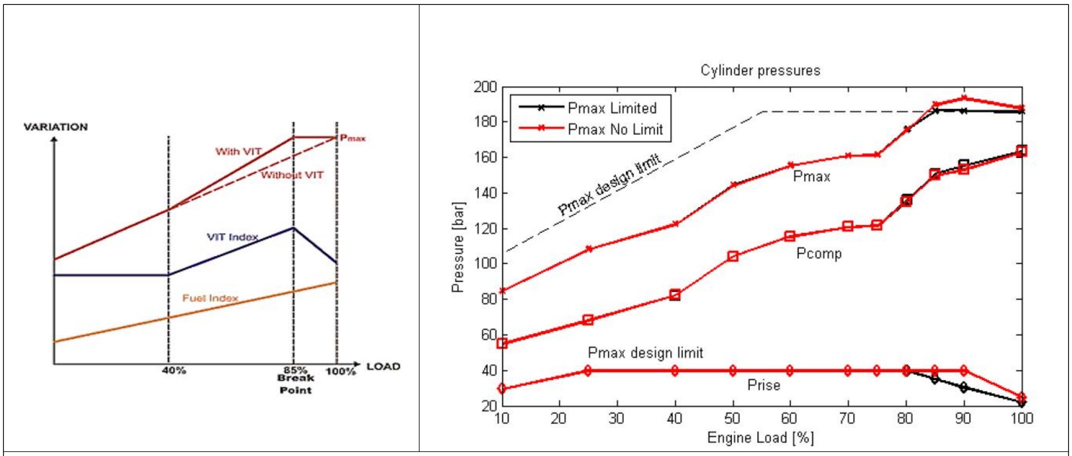


Fig. 3.11 Average Pressure Measurements Between Modes at NCR and MCR

It is clear from **Fig. 3.11** that while the engine is in gas mode, the engine does not experience any relatively significant losses in any of the pressure values shown evidencing that LNG is nearly just as effective as HFO. It should also be noted that the scavenging air pressure at MCR between diesel and gas mode was 2.79 bar and 2.69 bar, respectively, with a 3.65% difference showing that turbocharger efficiency has not been compromised because of the fuel type change.

An interesting observation made from the data in **Fig. 3.11** is that Pcomp in diesel mode increases approximately 11 bar from NCR to MCR. This is normal, however, while in gas mode, the Pcomp from NCR to MCR seems to plateau while it should steadily increase. This phenomenon of Pmax limiting and steady Pcomp raising as engine load increases is shown in **Fig. 3.12** and is complemented by a graph showing the effect of VIT on maximum cylinder pressure.



(Right-hand graph):
Dotted line: Pmax design limit example
Red line: Target performance based on optimal SFOC in site ambient condition.
Black line: Actual performance limited by Pmax design limiter

Fig. 3.12 Effect of VIT on Max Cylinder Pressure & Example of Pmax Design Limiter Functionality [28]

Under normal operations, as engine load increases above 40%, the start of injection advances in injection timing. When the engine load has reached about 85% the engine has normally reached Pmax and the VIT is automatically adjusted to retard injection timing in order to keep Pmax constant between 85-100% engine load. The operational data that was collected shows this effect on both modes in regards to Pmax, however, gas mode Pcomp pressures seem to plateau instead of rising during these engine loads.

Additional Pcomp and Pmax data from operational data collected onboard on December 2nd, 2017 and from the shop test results from the Hyundai Mipo Dockyard can be seen in **Fig. 3.13** and **Fig. 3.14**. From the operational data, it is obvious that in gas mode Pcomp seems to have a plateauing effect instead of steadily increasing as engine load increases as shown in the Pmax design limiter functionality example.

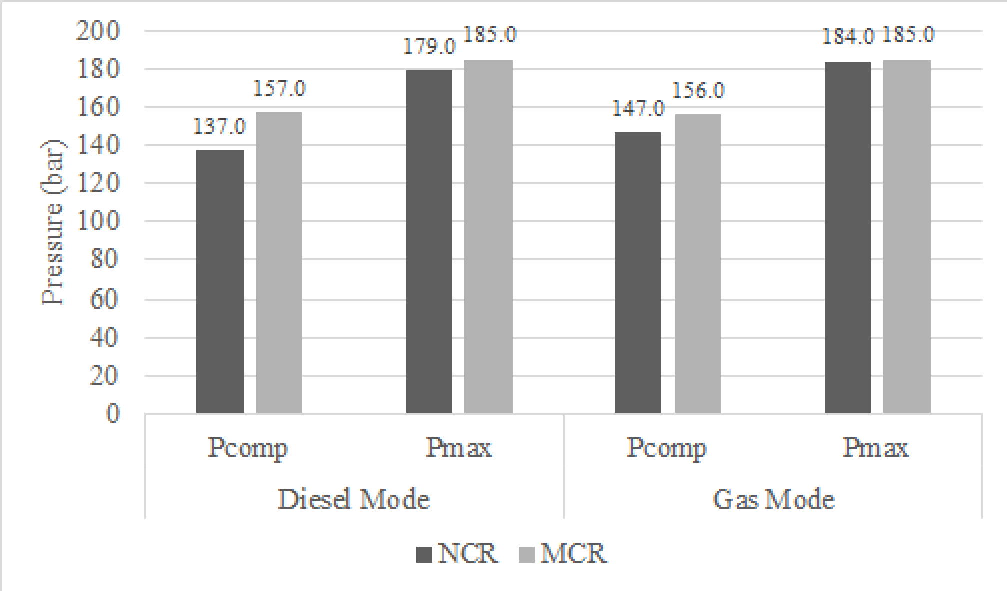


Fig. 3.13 Pcomp and Pmax Shop Test Results

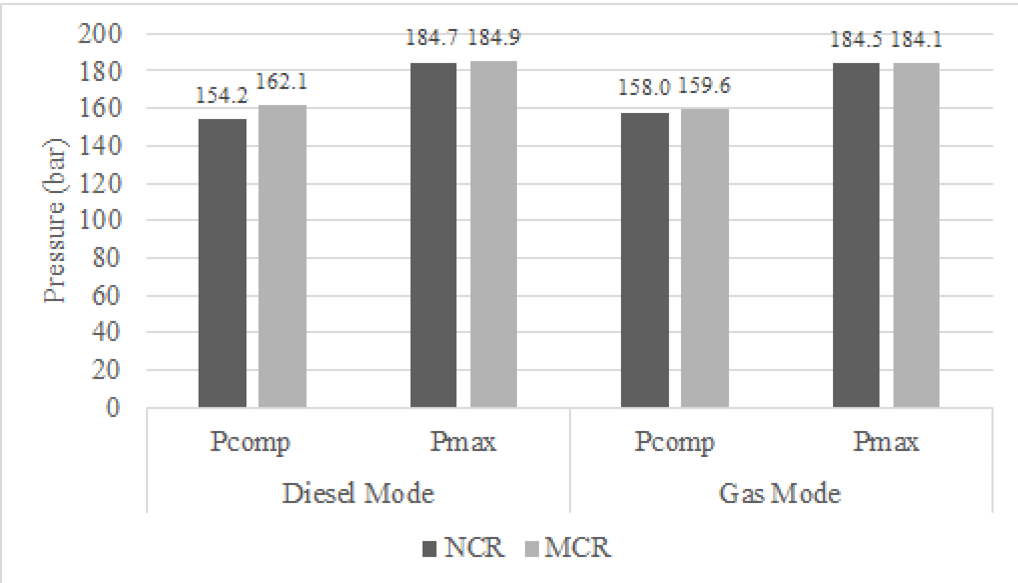


Fig. 3.14 Pcomp and Pmax Operational Data

Further investigation of this led to an inquiry with an official from the engine's manufacturer. An explanation as to why Pcomp and Pmax do not exhibit typical behavior at NCR through MCR was given by the official and is summarized as follows: Pcomp is determined by the scavenging air pressure and Pmax increases until the Pmax design limit is reached. Pcomp varies according to the scavenging air pressure, and the scavenging air pressure varies depending on the temperature of the atmosphere. Therefore, it is judged that the difference between the shop tests and sea trials of the gas mode Pcomp is due to the difference in atmospheric conditions. Sea trials conducted by the shipyard are carried out separately from the diesel and gas mode shop tests, which may result in different atmospheric conditions.

We can further demonstrate the validity of LNG as a practical alternative fuel by plotting the cylinder pressure against the cylinder's relative volume, or P-V% diagram. The resulting P-V% diagram data is also a product of the PMI measurements performed onboard and is shown in **Fig. 3.15** for both modes at NCR averaged across all six cylinders.

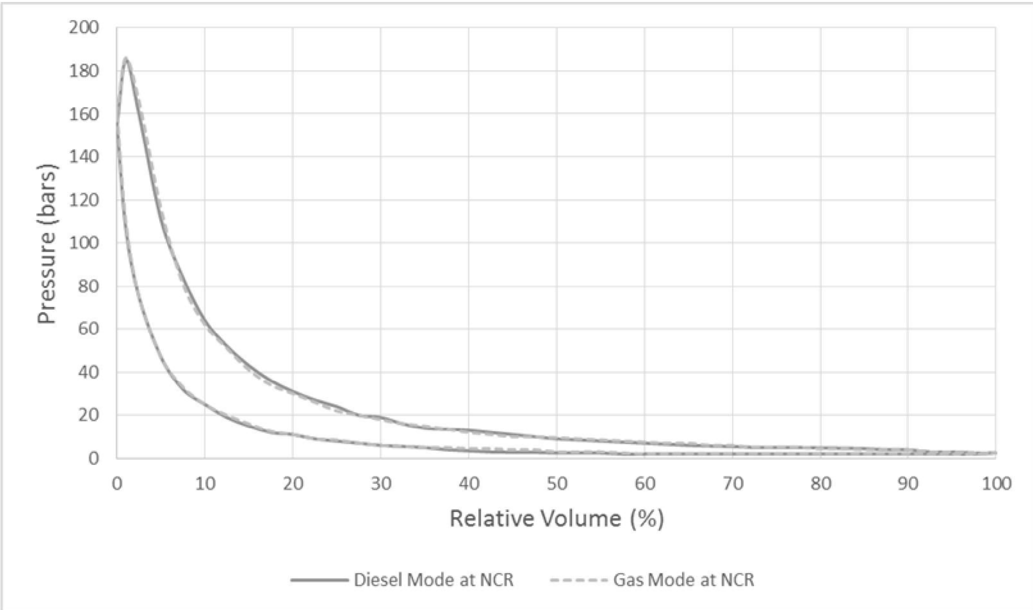


Fig. 3.15 P-V% Diagram for Diesel and Gas Mode at NCR

At NCR there is no relatively significant distinguishable difference between diesel mode and gas mode while analyzing the P-V% diagram. This demonstrates that gas mode has practically equal effective cylinder pressure compared to diesel mode validating no significant loss in total work performed by the engine. This attests to equally suitable engine performance compared to diesel mode.

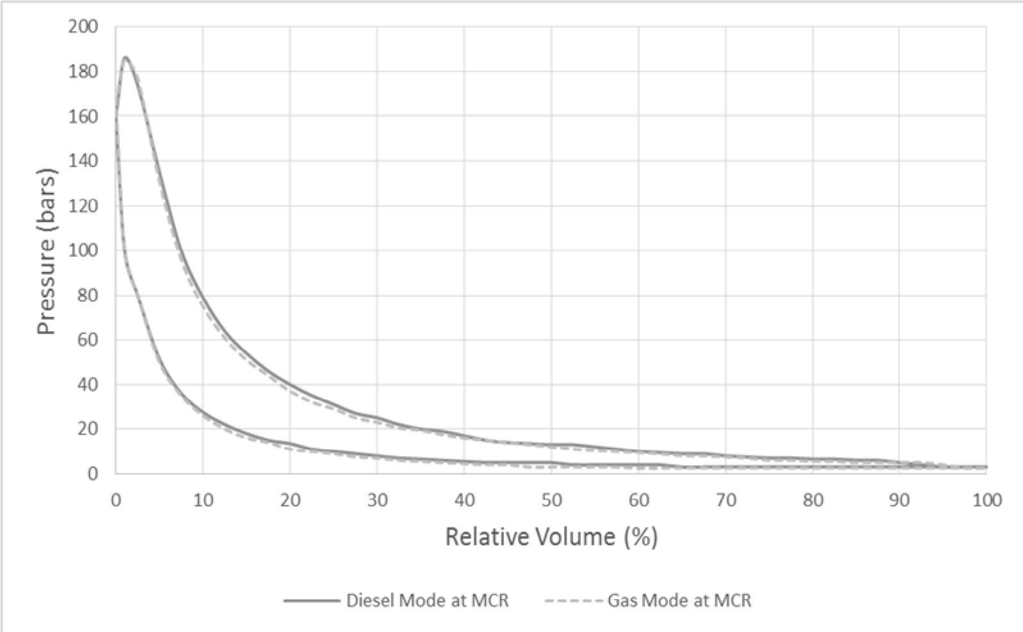


Fig. 3.16 P-V% Diagram for Diesel and Gas Mode at MCR

Fig. 3.16 shows the P-V% diagram for both modes at MCR. Once again, at MCR there is no relatively significant difference in cylinder pressures between modes. Having similar results with the NCR P-V% diagram, gas mode at MCR has nearly equivalent combustion efficiency as diesel mode.

An essential aspect while analyzing these PMI measurements is the fact that while the engine operates in gas mode, there is no significant difference in total effective pressure. This means that the power output of the engine meets the same power requirements of the shipowner while operating in either diesel mode or gas mode. This validates that LNG generates sufficient power to comply with the ship owner's power demand.

Chapter 4: Conclusion

This study analyzes engine performance of a ME-GI engine aboard the world's first 50,000 DWT LNG-powered bulk carrier, the MV Ilshin *Green Iris*. In order to verify that dual-fuel operation of LNG is a safe and efficient alternative method to traditional marine diesel combustion, gas and diesel mode, of the engine, are analyzed. Key results of the analysis are summarized as follows:

1. Transitioning between modes has a minimal effect on engine load relative to MCR as well as engine RPM and power output. Between modes, there is an average difference in power of about 0.85%, or approximately 61.68 kW, favoring diesel mode.

2. The IMEP for gas and diesel mode at NCR was calculated to be approximately 13.04 bar and 13.64 bar, respectively.

3. Gas mode has an overall higher thermal efficiency with an average percent difference of 1.91%, favoring gas mode. The thermal efficiency, at NCR and MCR, of diesel mode, is 50.6% and 49.8%, while gas mode is 51.4% and 50.3%, respectively.

4. Between modes, the SFOC has an average percent difference of about 16.27%, favoring gas mode, yielding an SFOC average difference of about 24.88 g/kWh.

5. Gas mode exhibits a lower exhaust gas temperature (before turbocharger) against engine load with an average difference of 10.17℃. Calculations also conclude that gas mode operation yields less exhaust gas with an average difference of 0.032 kg/s.

6. PMI analysis demonstrated that gas mode has practically equal effective cylinder pressure compared to diesel mode validating no significant loss in total work performed by the engine. There were, however, interesting results in gas mode in regards to Pcomp exhibiting an unusual plateauing effect from NCR to MCR.

The MV Ilshin *Green Iris* has served as a portal into understanding and validating that LNG is a safe, economically viable, and efficient fuel to be used by ship owners and operators, especially in this era of transitioning to alternative fuels. The results and findings of this study could lead to possible research in various topics such as mode transition's effect on engine behavior and performance, LNG combustion's effect on engine cylinder wear, and dual-fuel applications to other engine types, among others. This analysis has provided an unbiased, detailed look between the operational behavior of dual-fuel and diesel mode of high-pressure injection systems of large, 2-stroke marine propulsion engines. Contributions of this study to the maritime industry may help shape the future towards cleaner energy.

References

1. “Special Areas under MARPOL.” IMO, 2019, www.imo.org/en/OurWork/Environment/SpecialAreasUnderMARPOL/Pages.aspx.

2. Pedersen, Michael F. “International: IMO Marine Engine Regulations.” *Emission Standards: International: IMO Marine Engine Regulations*, Diesel Net, 2018, www.dieselnet.com/standards/inter/imo.php.

3. *IMO MEPC 72 Summary Report*. Lloyd's Register EMEA, 2018, *IMO MEPC 72 Summary Report*.

4. *ME-GI Dual Fuel MAN B&W Engines A Technical, Operational and Cost-Effective Solution for Ships Fuelled by Gas*. MAN Diesel & Turbo, 2012, www.corporate.man.eu/man/media/content_medien/doc/global_corporate_website_1/verantwortung_1/megatrends_2/klimawandel/me_gi_dual_fuel_en_01.pdf.

5. Dodge, Edward. “How Dangerous Is LNG?” *Breaking Energy*, Breaking Energy, 24 Dec. 2014, 12:00pm, breakingenergy.com/2014/12/22/how-dangerous-is-lng/.

6. Cho, Kwonhae, et al. *전용선 (Special Ship)*, Dasom, 2018, p. 169.

7. “LNG Safety.” *Bringing Energy Where You Need It.*, Prometheus Energy, www.prometheusenergy.com/services-solutions.

8. *2019 World LNG Report*. International Gas Union, 2019, www.igu.org/sites/default/files/node-news_item-field_file/IGU%20Annual%20Report%202019_23%20loresfinal.pdf.

9. Klein, Hermann J. “8th AVL Large Engines Tech Days 2018.” *Fuels and Power for Container Vessels*, 2018, p. 17.

10. *2017 World LNG Report*. International Gas Union, 2017, pp. 4-7, *2017 World LNG Report*.

11. Park, Moo-Hyun. “Kormarine Conference 2017 ‘Industry 4.0 and Green Business Creation.’”*Shipbuilding: Eco Is the New Drive on the Shipbuilding Industry*, p. 115.

12. “Energy.” *Bloomberg.com*, Bloomberg, 18 Apr. 2018, 4:46pm, www.bloomberg.com/energy.

13. “A Technical, Operational and Cost-Effective Solution for Ships.” Marine.mandieselturbo.com, 2014, marine.mandieselturbo.com/docs/librariesprovider6/technical-papers/me-gi.

14. “Benefits of LNG.” *WPCI*, World Ports Climate Initiative, 19 Apr. 2018, 5:31pm, www.lngbunkering.org/lng/environment/benefits-of-LNG.

15. *MAN B&W G70ME-C9.5-GI-TII Project Guide Electronically Controlled Dual Fuel Two-Stroke Engines*. 1st ed., MAN Diesel & Turbo, 2017, marine.mandieselturbo.com/applications/projectguides/2stroke/content/printed/G70ME-C9_5-GI.pdf.

16. *Fuels - Higher and Lower Calorific Values*, Engineering Toolbox, 2003, www.engineeringtoolbox.com/fuels-higher-calorific-values-d_169.html. 19 Apr. 2018, 9:08pm.

17. Baijal, Nalin. “Engine Timing and Terminology.” *Basics of Marine Engineering*, 1 Jan. 1970, basicsofmarineengineering.blogspot.com/2013/01/engine-timing-and-terminology.

18. Rowen, Alan L, and R. D. Jacobs. “Marine Diesel Engines.” *Modern Marine Engineer's Manual*, by Everett C. Hunt, 3rd ed., vol. 2, Cornell Maritime Press, 2002, p. 48.

19. “Dual- Fuel Engines From Wärtsilä.” *Wärtsilä*, 2 May 2018, 3:37pm, www.wartsila.com/encyclopedia/term/dual--fuel-engines-from-wC3%A4rtsil%C3%A4.

20. “ME-GI Engine Fueled by LNG.” *Lngbunkering*, 2011, www.lngbunkering.org/lng/sites/default/files/2011%20MAN%20ME-GI%20engine%20fuelled%20by%20LNG.pdf.

21. Cantu, Chris. “Study Looks at Emissions, Economic Characteristics of Dual-Fuel, High-Horsepower Engine Used in Hydraulic Fracturing Application.” *Drilling Contractor*, 27 Apr. 2016, www.drillingcontractor.org/study-looks-emissions-economic-characteristics-dual-fuel-high-horsepower-engine-used-hydraulic-fracturing-application-39231.

22. Ashok, B. “LPG Diesel Dual Fuel Engine ‑ A Critical Review.” *ScienceDirect.com | Science, Health and Medical Journals, Full Text Articles and Books.*, Science Direct, June 2015, www.sciencedirect.com/.

23. Latarche, Malcolm. “Dual Fuel and Gas Engines.” *ShipInsight*, Ship Insight, 31 Dec. 2017, 4:02pm, shipinsight.com/dual-fuel-gas-engines/.

24. Wang, Hao. “Feedback Control during Mode Transition for a Marine Dual Fuel Engine.”*ScienceDirect.com | Science, Health and Medical Journals, Full Text Articles and Books.*, Science Direct, 2015, www.sciencedirect.com/.

25. Thomas, et al. “BACKGROUND.” *DUAL-FUEL ENGINE COMBUSTION MODE TRANSITION CONTROLS*, FPO, 30 Mar. 2017, www.freepatentsonline.com/y2017/0089273.html.

26. Corkhill, Mike. “POSCO High-Manganese Steel Debuts in LNG Bunker Tank.” *LNG World Shipping*, 22 Feb. 2018, www.lngworldshipping.com/news/view,posco-highmanganese-steel-debuts-in-lng-bunker-tank_50885.htm.

27. *Instructions For 46-98 MC Type Engines Operation*. 40th ed., MAN B&W Diesel A/S, Copenhagen, Denmark, 1998, p. 176, 180.

28. “The MAN B&W MC Engine VIT Fuel Pump.” *Marinediesels.co.uk*, www.marinediesels.info/2_stroke_engine_parts/Other_info/MANBW_VIT_pump.htm. & MDT Technical Project Presentation.

Acknowledgments

It is an honor for me to first thank both my thesis advisors Professor Kangki Lee and Professor Kwon Hae Cho of the Korea Maritime and Ocean University. My advisors both gave me great inspiration through lectures and personal meetings. They were always there whenever I had questions about my research and never hesitated to meet with me. The time and dedication they gave to me were immeasurable. The professors consistently guided me in the right direction whenever I needed it through their invaluable insight and mentorship.

I would also like to thank the many incredibly talented professors at the university. Their willingness to help and teach gave me enormous encouragement and their seemingly boundless knowledge stimulated my curiosity to no end. I would also like to give my gratitude to Professor Sihwa Kim, who initially opened the door to my education at the university and greatly inspired me to pursue my master's degree. I am also indebted to my fellow colleagues, classmates, and friends who supported me, particularly to Dahee Lee, whose help and assistance through my time at the university is deeply appreciated.

I owe my most profound gratitude to my parents, whose continuous and unfailing support over the years of my studies has provided me with a solid rock of encouragement. I would like to thank my two sisters, Mallory and Lindsay, who also inspired me to pursue higher education. And finally, I would like to express my warmest gratitude to my lovely wife, Jihye, who supported and encouraged me through this endeavor every day and whose tremendous help guided me to the very end. There is not enough paper in this thesis to express the gratefulness I have for everyone who helped me. This accomplishment would not have been possible at all without them.

Thank you.